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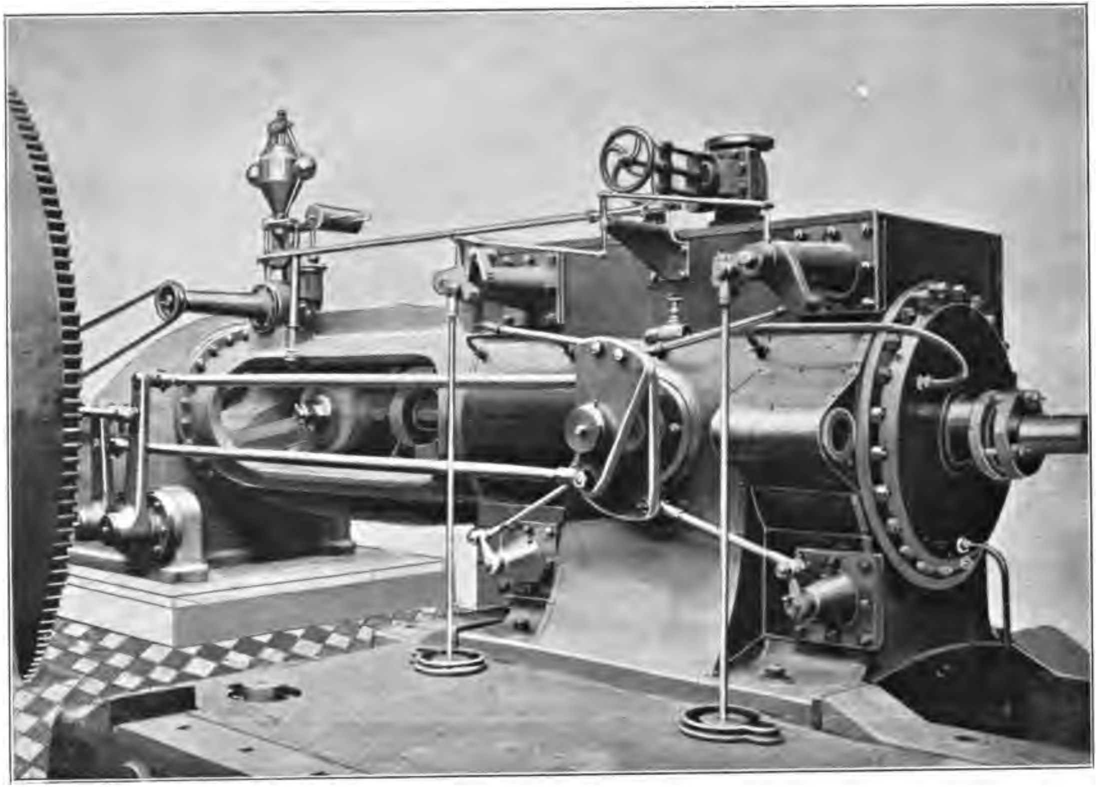

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PREFACE TO THE THIRD EDITION.

IN this edition a Chapter on Drop Valve Gears has been added, and the original portion of the work has been revised.

BURTON JOYCE,
NOTTS, *November*, 1902.

PREFACE.

In the preparation of this work it has been my aim, as far as possible, to deal with the subject of Valve Gearing in a simple and interesting manner.

Whilst in the main adopting the Zeuner diagram, as being the most suitable for determining the proportions of Valve Gear, I have not confined myself exclusively to that method of solution; my object being to show that valve movements may be described in various ways, each possessing its own advantages.

In the First Part, chapter I. treats of the common Slide Valve and its modifications; the second chapter is devoted to Expansion Valves and Automatic Cut-off Gearing; and the remaining chapters deal with Link Motions and other Reversing Gears.

The Second Part treats of Corliss Valves and Trip Gears, a branch of the subject which has hitherto received but ittle attention from writers on Valve Gearing.

I have to acknowledge my indebtedness to several eminent firms who kindly furnished examples of the best modern gears; and I have also to thank my friend, Mr. F. H. Parker, for his assistance during the preparation of the work for press.

CHARLES HURST.

WIGAN, *June, 1897.*

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VALVES AND VALVE-GEARING.

PART I.

SLIDE VALVES.

CHAPTER I.

SIMPLE VALVE GEAR.

IN dealing with valve gears, the common slide valve suggests itself as being a suitable starting point, for once the simple diagrams and requirements relating to it are understood, the subject leads easily and naturally to the consideration of more complex gears.

Area of Ports.—The first consideration when designing any valve gear is the amount of steam and exhaust openings requisite for the free passage of steam. It has been found that the velocity of steam entering the cylinder should not exceed 6,000 to 7,800 feet per minute; while that of the exhaust should not exceed 4,800 to 6,800 feet per minute.

Let A = area of port opening in square inches.
D = diameter of cylinder in inches.
P = piston speed in feet per minute.
V = mean velocity of steam in feet per minute.
V^1 = mean velocity of exhaust in feet per minute.

Then

$$A = \frac{D^2 \frac{\pi}{4} \times P}{V}.$$

The area of opening for exhaust will be—

$$\frac{D^2 \frac{\pi}{4} \times P}{V^1}.$$

In deciding upon the values of V and V^1, regard must be had to the length and shape of the ports, and to the intended pressure of steam. In long-stroke slide valve engines the lesser velocities should be taken; but in Corliss engines, where the ports are short and direct, the higher values of V and V^1 may be taken without fear of wiredrawing. The frictional loss of steam in the ports is proportional to its density; hence it is necessary to consider the pressure when calculating the area of passages. An instance of the effect of this consideration may be found in a well-designed compound engine. In the high-pressure cylinder the ports may have such an area as will give a velocity of, say, 5,700 feet per minute to the steam and 5,200 to exhaust; but in the low-pressure cylinder it is not uncommon to find these velocities to be 7,200 and 6,500 respectively.

Having determined the area of the ports, the length must be fixed upon to find the width; or, if the width be settled, the length is easily found. It is to be noted that the steam port should be equal in area to the amount given by the formula for exhaust—that is, in slide valve engines—as it is evident that the exhaust must pass through the steam port. Allowance sho ld be made for any ribs which may be cast in the ports, as they contract the passages.

Lead.—In fixing the amount of lead, the chief point requiring attention is the piston speed. Other things being equal, the lead increases as the piston speed. Large clearance engines require more lead than Corliss engines. A common practice is to allow in medium-stroke engines $\frac{1}{32}$ inch linear lead for every 100 feet of piston velocity. In vertical engines the lead on the top side of the cylinder is less than on the bottom, for two reasons. Firstly, because gravity is against the piston, piston-rod, crosshead, and connecting-rod on the up-stroke; and secondly, because most of the wear in the joints of the link is in a downward direction. It is a common practice, when first setting vertical engines to work, to give on the top side half the lead given on the bottom end.

Point of Cut-off.—The percentage of cut-off with a common slide valve may be from $\frac{5}{8}$ to $\frac{7}{8}$ of the stroke. Greater expansion than the earlier cut-off given should be obtained by expansion valves; with a later cut-off than $\frac{7}{8}$ there is likely to be excessive back pressure. A cut-off at 75 per cent. is very convenient; the exhaust and compression then being quite satisfactory, and the travel of the valve not abnormal.

Having decided upon these points—namely, the port opening, the lead, and the percentage of cut-off—the design of the valve may be at once proceeded with.

Diagram of Simple Valve.—On a horizontal line A B (Fig. 1),

draw C D for the position of the crank when steam shall be cut-off. From C, make C E equal to the lead, and draw E F perpendicular to A B. Draw B G parallel to E F, making E B equal to the required width of steam opening. Next, draw C H at right angles to C D, cutting the line E F at J. Bisect H J F by line J K. On J K find the centre of a circle which shall pass through C and touch the line B G. Let L be the centre thus found. Then, in the diagram, L C or L G equals the throw of the eccentric, or half the travel of the

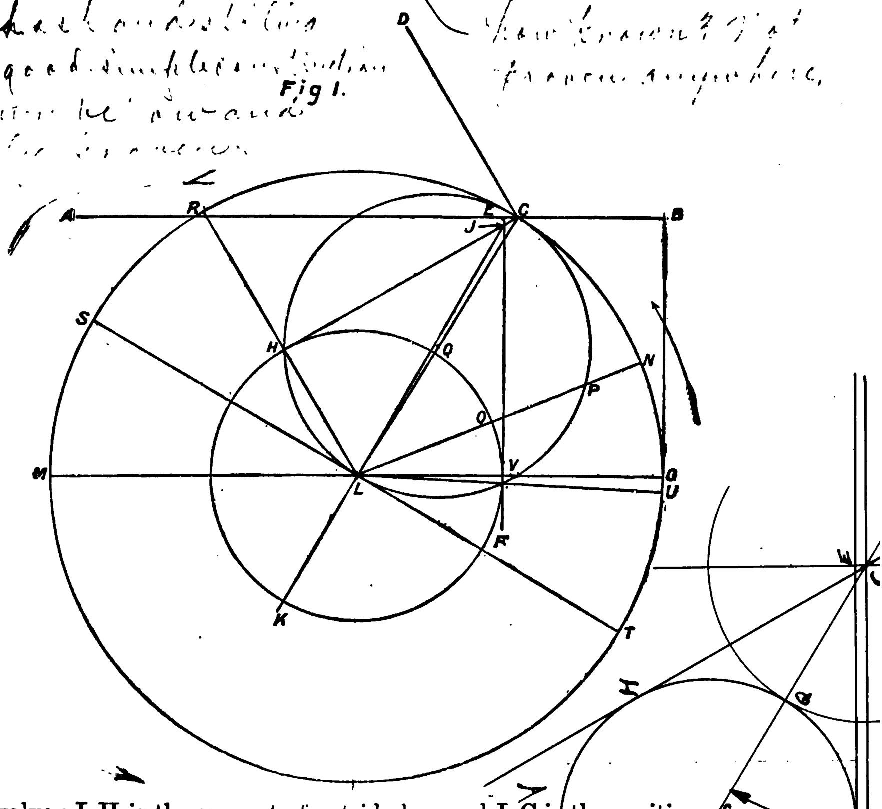


Fig 1.

valve; L H is the amount of outside lap, and L C is the position of the eccentric sheave relative to the position L M of the crank, L M being parallel to A B. On L C describe the primary valve circle C H L. This circle cuts the lap circle H K in H. Draw the line L H R. This is the position of the crank when cut-off occurs. The line L R should be parallel to C D; and the fulfilment of this condition is a test for the accuracy of the diagram.

That this construction is correct is proved by the following reasoning:—The line H C is drawn at right angles to D C; and in

LEAD = CE
PORT OPEN'G = CQ
ST. LAP = LQ
CR. POS. FOR C-O-CH
ECC'Y = CL
ANG. ADV = δ

the semicircle L H C, the angle C H L is a right angle (Euclid's Elements, Bk. iii., Prop. 31); L R is therefore at right angles to H C, and, consequently, parallel to D C.

Again, because the circle H K O is drawn so that the line H C is tangent, the line L R is at right angles to that tangent (Euclid, Bk. iii., Prop. 18), and therefore parallel to the given line of cut-off D C.

That the maximum port opening is equal to the given amount E B can readily be shown. The line E F is drawn at right angles to A B, and so is line B G. Therefore, E F and B G are parallel. The line E F is also tangent to the circle K H O, by construction; and, therefore, since the half travel of a slide valve is occupied by lap and port opening, the distance L G is equal to the port opening *plus* the lap. But it has been shown that V G is equal to the port opening; therefore L V is the lap.

The whole principle of the Zeuner diagram is directly founded on the 31st proposition of Euclid's third book; and the truth of all diagrams can be proved by reasoning similar to that given above.

Taking any position of the crank, as at L N, the condition of things is clearly shown by the diagram. The port will be open to steam an amount equal to O P, given by the distance between the lap circle and the valve circle on the line of the then position of the crank. Also, at this position, the valve is distant to the left of its central position an amount equal to L P. At crank position L O the valve is at its greatest distance to the left of its central position, the port opening being equal to Q C; Q C being equal to E B as arranged for. Again, at L R the valve has just closed the port; L H being its displacement to the left of its mid position. At L S the valve has arrived at its central position and travels to the right until L K is reached. It now returns and arrives at its central position at L T; the line S T being at right angles to L C. At L U the port is open ready for the commencement of the stroke.

Thus far only the action of the entering steam has been considered. In order to study the exhaust Fig. 2 is constructed. In actual working this diagram would not be drawn separately, but it is to facilitate explanation and avoid confusion that two diagrams have been drawn in this instance.

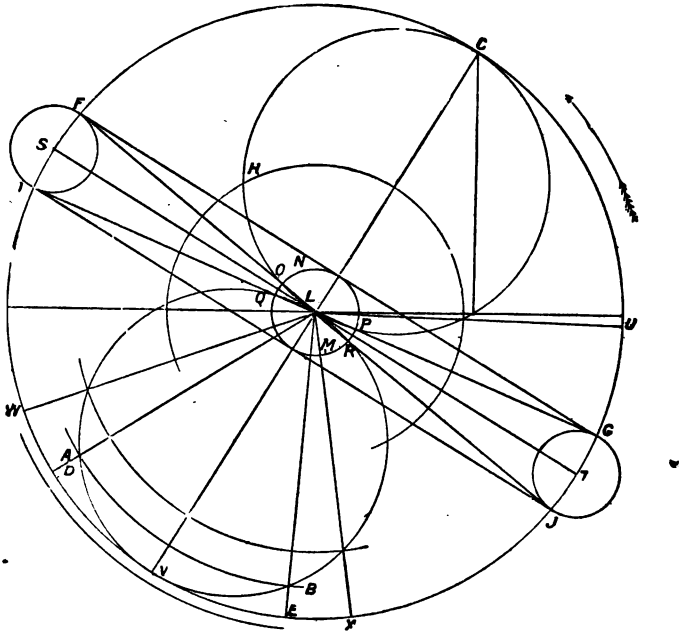
Make L C, L H, and angle C L U, respectively, equal to L C, L H, and angle C L U in the first diagram. Produce C L to V, and on L V describe the secondary valve circle. Now, as the primary valve circle gave the port openings to steam for any crank position from T to S, so will the secondary valve circle give the port openings and valve positions for any point of the stroke from S to T, in the direction of the arrow.

Taking a case in which there is no inside lap, at crank position L S, the valve is in its central position, as previously explained; and it is evident that when it is in this position the valve is about to open for exhaust either one port or the other. L S, therefore, represents the position of the crank when release occurs. The port is being opened from S to V. Of course, the opening cannot

exceed the width of the steam port, so that, if from L as centre, an arc A B be struck, the radius of which is equal to the width of the steam port, it is clear that the maximum port opening is attained at crank position L D, and this opening is maintained till L E is reached. At this point reduction commences and continues till L T. At L T the port is closed, and the steam remaining in the ports and cylinder is compressed until admission occurs at L U.

Considering now the effect of inside lap, it will be observed that positive inside lap delays release and hastens compression; while negative lap has the reverse effect in an equal degree. From L as

Fig. 2.

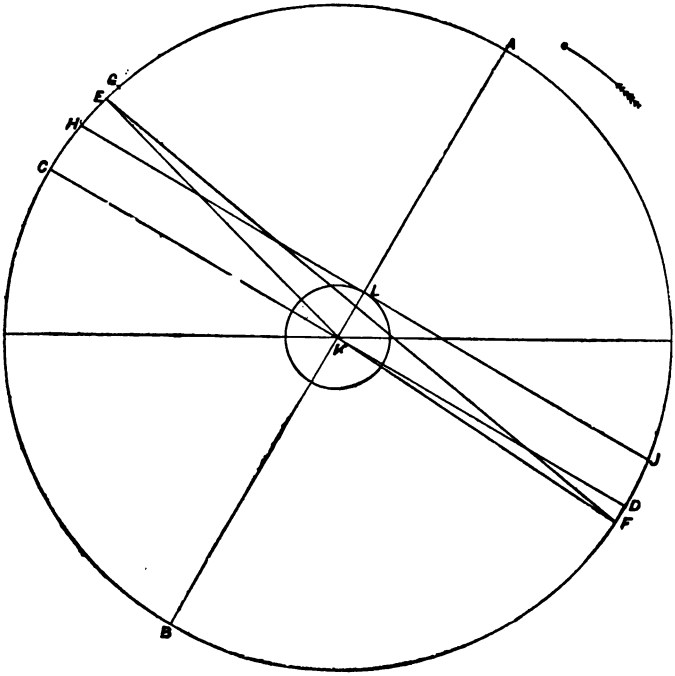


centre, and with radius equal to the inside lap, either negative or positive, describe the circle M N. This circle cuts the primary valve circle at the points O and P, and the secondary circle at Q and R. F om L draw lines pas-ing through these points, and produce them to the circle C D E. Then, with positive lap, L I is the position of the crank at release, and L J its position at compression. With negative lap, L F is the release position, and L G compression position. The same result may be obtained by a different method. On S and T as centres, and with radius equal to M N (the inside lap), describe the circles I F and G J. These circles cut the valve travel circle at points F G I and J, and these are the required points.

In practice, however, it would usually happen that the points of release and compression would be decided upon. Then it would be necessary to obtain the inside lap to effect the desired result. Let F and G be the points decided upon. Join F and G by the straight line F N G. Then, if this line be parallel to S T, L N is the required negative inside lap; the circle M N having F N G as a tangent. If I and J were the points chosen, L M would be the necessary positive inside lap, the line I J being parallel to S T.

This construction only applies to cases where the line joining the two chosen points is at right angles to the centre line of the valve

Fig 3.



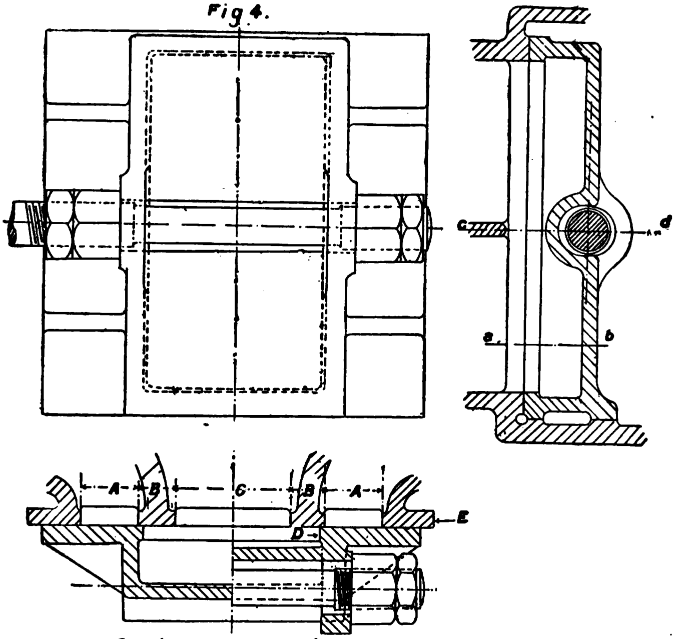
circles; and when this is not the case it is to be concluded that no amount of lap, neither negative nor positive, will effect the required distribution; but it is easy to arrange matters so that the resulting distribution shall be a compromise between the two points. In the figure above, A B represents the centre line of the valve circles, and C D is at right angles to A B. E and F are the chosen points for release and compression. But the line E F is not parallel to C D; therefore this distribution is not possible From E make E G equal to F D. Bisect the arc C G in H, and draw H J parallel to C D. If the lap be made equal to K L, release and compression will be at

H and J respectively; and these points will be the nearest approach to the required distribution that it is possible to attain. When the line joining the two given crank positions intersects the primary valve circle the lap will be negative; and when that line intersects the secondary valve circle the lap will be positive; and when the line is at right angles to the centre line of the valve circles there will be no inside lap whatever.

In Fig. 2, if there be negative lap equal to L N, the port will be fully open to exhaust at crank position L W; the arc, whose radius equals the width of the steam port minus negative lap, being struck from L as centre. L X is the point at which reduction commences. With the same amount of positive lap, it is evident that the port will never be fully open to exhaust, as shown by the arc falling without the valve travel circle.

Percentages of Release and Compression.—In determining the most suitable point for compression, the clearance is the chief

Fig 4.

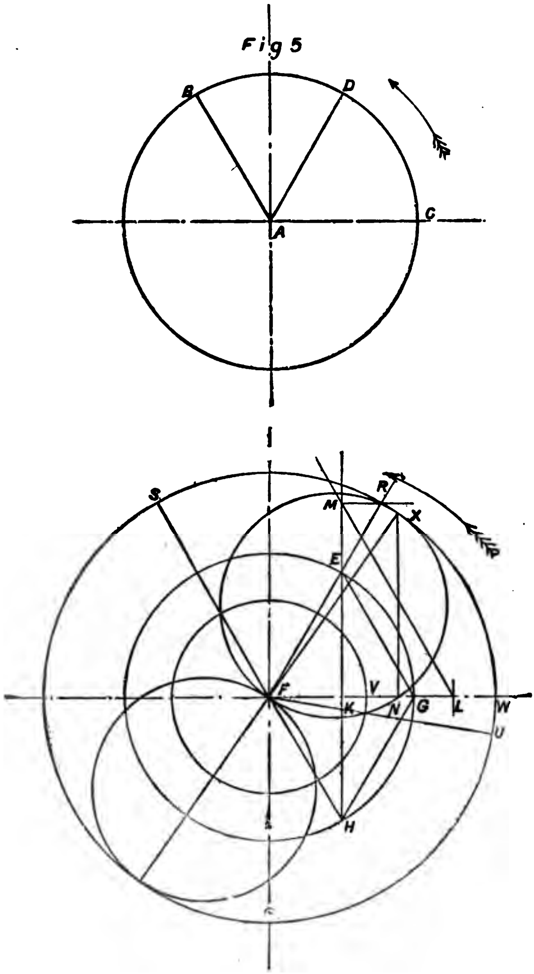


Section at a.b. Section at c d.

point to consider. The most satisfactory way is to plot down the probable indicator diagram, and by means of the hyperbolic curve, to find a compression that will give a final pressure in the cylinder to balance the momentum of the moving parts. Having settled on this point, the valve diagram is drawn, and the release point finds itself. It is possible that release may fall before cut-off, in which

case less compression must suffice in order to cause release to fall later in the stroke. Release at 92 per cent., and compression at 89 per cent., have been found very satisfactory for horizontal engines, with piston speeds of from 450 to 600 feet per minute,

Fig 5



and with clearance volumes of from 5 to 10 per cent. of the piston's displacement.

In high-speed vertical engines, it is the practice to give more compression on the under side of the piston, in order to counteract gravity. A compression at 82 per cent. on the bottom side is not uncommon in such cases.

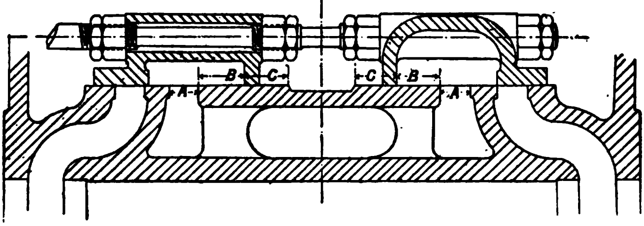
Application of the Diagram.—The practical application of the valve diagram is illustrated by Fig. 4. A is the width of the steam port as obtained by calculation. B may be any convenient dimension. Then C should equal distance L C (*i.e.*, half travel of valve, see Figs. 1 and 2) + A (Fig. 4) + L; L being *plus* for positive inside lap, and *minus* for negative inside lap. When the port and valve faces are thus designed, the port opening through C will never be less than the width of the steam port; that is, assuming that the width of the bar, B, is less than L C − L when there is positive lap; and L C + L when there is negative lap. The edge of the valve, D, must on no account slide past the outer edge of the port face, E.

The following is a very useful extension of the Zeuner diagram for solving the foregoing problem:—

Let A B in the upper diagram of Fig. 5 represent the position of the crank when it is desired to cut off. Bisect the angle B A C by the line A D. In the lower diagram, draw the circle E H equal to he assumed circle B D C above. Make angles G F E and G F H both equal to angle D A B or D A C; and join E and H by the line crossing the horizontal at K. From K, lay off K L equal to the required maximum port opening *minus* half the lead the valve is to have. Join G E, and draw L M parallel to G E, cutting H E produced, in M. From M draw the horizontal M R, intersecting F E produced, in R. Then F R is the half travel of the valve. Draw the circle R S W. From W, lay off the maximum port opening W V, which discovers the lap F V. Draw the lap circle; and make V N equal the given lead. Draw the perpendicular N X. Then F X is the eccentric position when the crank is on dead centre; and drawing the valve circle having F X as a diameter, will show the steam action. In the figure, admission is seen to be at F V, and cut-off at F S, and F S is parallel to A B above.

Divided Valves.—When the stroke of an engine is very long in proportion to the diameter of the cylinder, the clearance with a

Fig. 6.



slide valve is great. This clearance, however, may be considerably reduced by placing a valve at each end of the cylinder. Fig. 6 shows such an arrangement. The section of the front valve is taken

through the boss for spindle; whilst the back is shown by a section above the spindle, and shows the dish-shaped form of the valve.

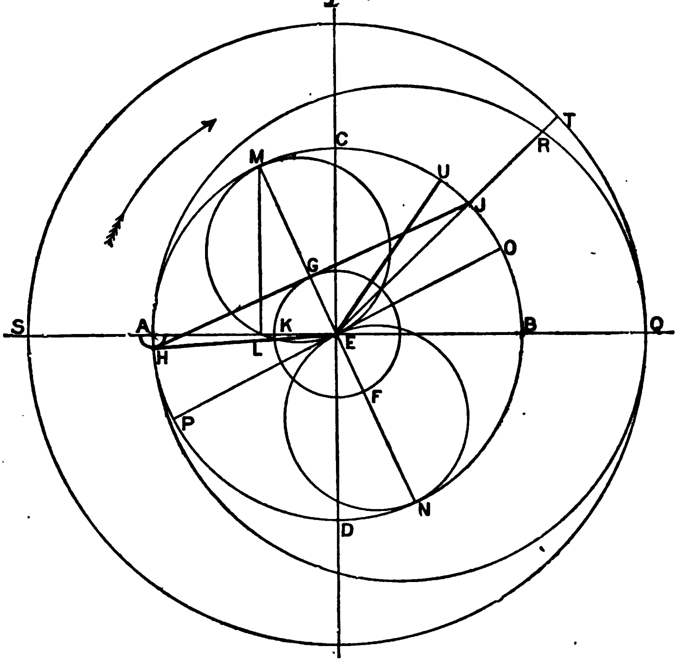
The proportions of the port and valve faces are here given.

A = width of steam port.
B = half travel of valve.
C = half travel of valve + ½ inch to 1 inch.

A case may frequently arise in which the travel of the valve, lap, and lead are known, and it is required to find the points of cut-off, release, and compression. This problem presents itself when the actions of existing valves are being considered, so that alterations can be made if the steam distribution be found unsatisfactory.

Valve Diagram.—On A B (Fig. 7) draw the circle A B C D, equal in diameter to the given travel of the valve. From centre

Fig 7.



E, and with radius equal to the given outside lap, describe the circle F G. From centre A, and with radius equal to the given lead, describe the circle. The line H J drawn as a tangent to the lap and lead circles cuts the circle A B C D in H and J. E H is the position of the crank when admission takes place, and E J is the crank position at cut-off. From K, set off K L equal to the lead,

and erect a perpendicular L M. Join E and M. Then E M is the position of the eccentric sheave relative to the position of the crank E B. Produce M E to N, and on line M N describe the primary and secondary valve circles. The completion of the diagram is now a simple matter, and it will suffice to say that E O is release point, and E P compression point, the valve having no inside lap. The arrow indicates the direction of rotation, or rather the direction in which the steam distribution is traced; and although the eccentric is spoken of as being at E M, it should be remembered that this is its position relative to the real position of the crank E B, as said. To show the eccentric relative to the crank when at A E, the line E U is drawn; angle C E U being equal to angle C E M.

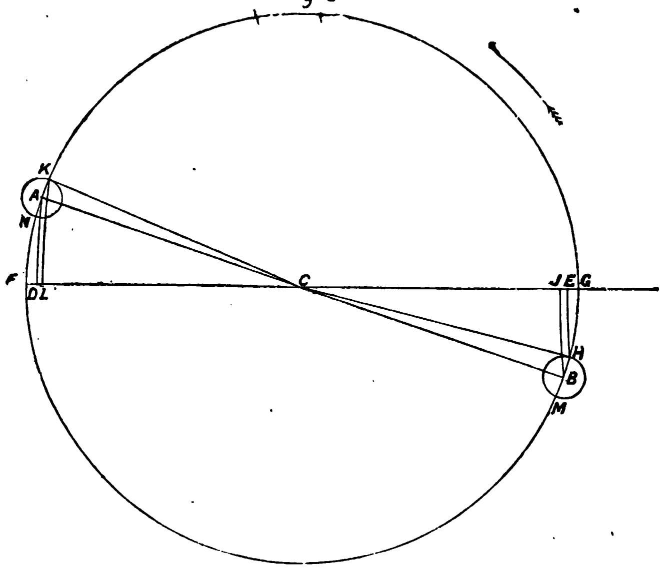
In all cases connected with the ordinary slide valve, it is well to remember that the crank being on dead centre, the centre of the eccentric sheave is in advance of the crank by 90 degrees *plus* the angle necessary for lap and lead; and further, if the valve had neither lap nor lead, the eccentric sheave would be at right angles to the crank and leading.

If, to any convenient scale, B Q be made equal to the stroke of the piston, the distance of the piston from the beginning of the stroke for any crank position is given by the distance between the valve travel circle A B C, and the circle Q R A, on the line indicating that position. Thus, at crank position E J, J R is the distance of the piston from the beginning of the stroke. Similarly, the circle Q T S gives the distance of the piston from the end of the stroke for any crank position. At crank position E J, R T is the distance of the piston from the end of the stroke. J R + R T = B Q, wherever J R and R T may be.

Effect of Connecting-Rod and Equalisation of Steam Distribution.—In all previous diagrams the angularity of the connecting-rod has been neglected; but in most cases it is necessary to take this angularity into account. The Zeuner diagram admits of this, but not of that of the eccentric-rod; the ratio of the length of the eccentric-rod to its travel, however, being in most cases great may be neglected. The effect of connecting-rod angularity is to cause events in the forward stroke to occur later than the corresponding events in the back stroke. Thus, a connecting-rod of five cranks in length would cause a cut-off at 75 per cent. in the forward stroke to occur at 68·75 per cent. in the back stroke; and with a shorter rod the difference would be greater. It is possible to equalise these events. It is also possible to equalise release and compression; and Fig. 8 shows how this may be done. Let C A be the crank position at release for the forward stroke. It is required to cause release in the back stroke to occur at the same fraction as in the front stroke. Produce the line F G, and with a centre somewhere on this line describe the arc A D, the radius of which bears the same ratio to distance A C, as the connecting-rod bears to the crank. F G representing the stroke of the engine, it is evident that the piston has travelled a distance G D from the

beginning of the stroke when release occurs; and A C is the corresponding crank position for this point. From G, make G E equal to F D. With centre on F G produced draw the arc E H. Produce C A to B. The distance B H will represent the difference between the inside lap of the front end of the valve and the inside lap of the back end. For example, suppose that in order to release at A C in the front stroke no inside lap were required; then the back end of the valve would have no lap, for it is the back end of the valve which controls the forward stroke. But if release is to occur at C H in the backward stroke, the front end of the valve must have positive inside lap equal to B H, because the front end controls the backward stroke.

Fig 8

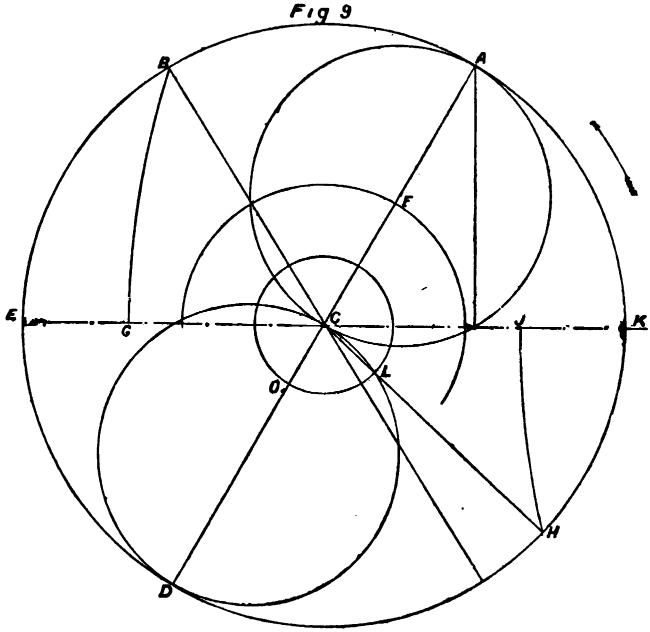


Dealing now with compression, it will be observed that if C B represent the crank position when this event takes place, the piston will be distance G J from the end of the backward stroke; F G representing, as before, the whole stroke. From F, make F L equal to G J, and draw the arc L K. Produce C B to A. Then A K will be the difference between the inside laps of the valve in order to equalise compression. In the present instance B H = A K; therefore both compression and release are equalised.

It has been mentioned that cut-off may be equalised. Fig. 9 shows how this may be effected:—C B is the position of the crank at cut-off, in the forward stroke. A C is the centre line of the eccentric. Produce A C to D; and on A D describe the valve circles. C F is the lap circle for the back end of the valve. With

the length of the connecting-rod as radius, draw the arc B G. Make J K equal to E G, and draw the arc H J. Join C and H. Where C H crosses the secondary valve circle (this circle being really the primary circle for the back stroke) describe the circle L O, the radius of which will be the necessary outside lap for the front end of the valve. Equalisation of cut-off has been obtained at the expense of unequal port openings (F A being the amount for the back port, and O D the amount for front port) and unequal leads. This is a serious objection; hence, equalisation of cut-off with the common slide valve is never attempted in practice.

Fig 9



Ellipse Diagram.—A convenient way of studying the action of the slide valve is to plot the motion of the piston as abscissa, and that of the valve as ordinate. On a horizontal line (Fig. 10) draw the path of the crank pin to any convenient scale, and divide it into any number of equal parts. On the horizontal line, and to any convenient scale, draw the eccentric circle. Make the angle G *c* *a*′ equal to the angular advance of the eccentric. From *a*′ divide the circle into equal parts to correspond with the crank-pin circle. Draw the arcs for the connecting-rod and the eccentric-rod, the centre line of the eccentric-rod, A B, being at right angles to the centre line of the crank-pin circle, A G. Starting at point *a* on the crank-pin circle, erect a perpendicular *a* 1. *a*′ is the corresponding eccentric position, and D will be the valve position. Project D 1

horizontally, meeting the vertical *a* 1 at 1. The first point in the curve has now been obtained. Repeat the process for each position, and draw a fair curve through all the points thus found. This curve is called the valve ellipse. Draw E F parallel to *a* G, and

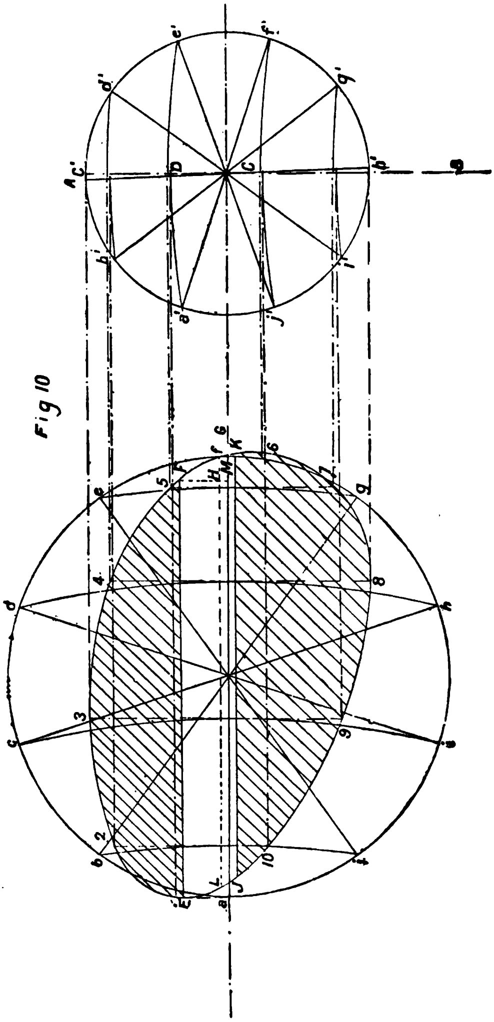
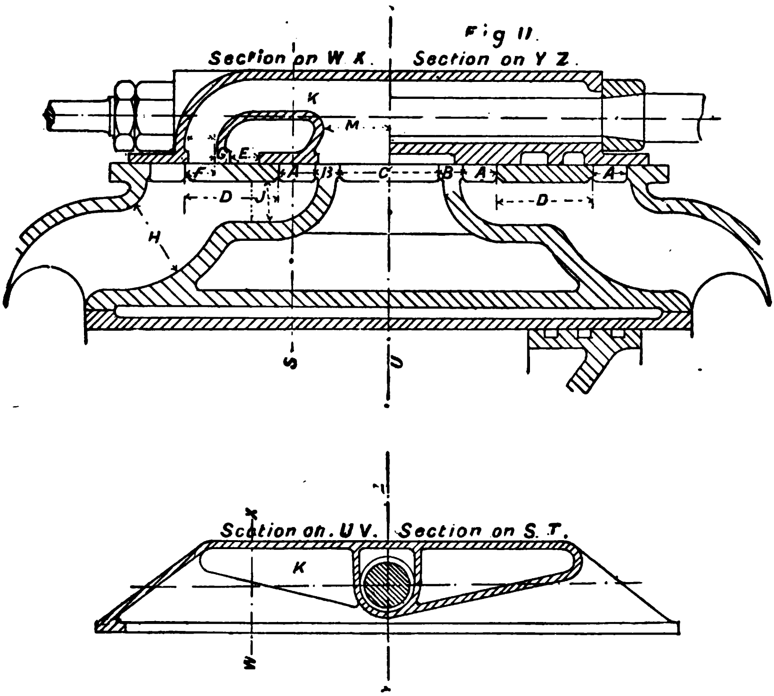


Fig 10

remote a distance equal to the outside lap. This line cuts the valve ellipse at points E and F. E is the point at which admission occurs; and F is the point of cut-off. Projecting these points to the line *a* G, which represents the stroke of the piston, their per-

centages may be easily obtained. When the piston is at *a*—that is, the beginning of the stroke—the valve will be distant, *a* E, from its mid position. Again, at piston position, H—the projection of F—H F is the displacement of the valve from its central position, and the port is just closed to steam. If J K be parallel to *a* G, and remote a distance equal to the positive inside lap, K and J will be the points of release and compression respectively. For negative inside lap, L M would be drawn above *a* G; and L and M would be the points of compression and release respectively. The upper shaded portion of the figure will give the port openings to steam for any crank position; whilst the lower shaded portion illustrates the opening to exhaust.

It is evident that this diagram is not suitable when designing new valves, but for tracing the action of existing gear, where the travel of the valve and the angle of advance are known, it is very convenient, inasmuch as it permits of both connecting-rod and eccentric-rod being taken into account. This is the only advantage over Zeuner's diagram.



Double-Ported Valve.

Double-Ported Valves.—Cylinders of large size require ports of considerable width; and consequently, the necessary travel of

the valve becomes excessive. Other things being equal, the work required to actuate the valve varies directly as the travel; hence it is desirable to make the travel as small as possible. The double-ported valve gives twice the steam opening of an ordinary valve with the same travel, and is much used for large cylinders. The valve diagram is drawn exactly as for an ordinary valve. The port openings and lead, however, will be twice the amount shown by the diagram. An example of a double-ported valve is given in Fig. 11. The correct proportions of the port and valve faces are here given :—

A = half total width of required exhaust opening.
C = 2 A + half travel of valve ± inside lap (+ L for positive lap, – for neg.) – B.
E = half total width of required steam opening.
F = half travel of valve.
D = E + G + F + outside lap.
J = A.
H = 2 A.

Area of passage K = area of one steam port.
M = not less than A ; preferably more.
B and G, of course, are empirical.

The figure shows a valve suitable for a vertical engine. The spindle is prolonged to carry a balance piston at the top of the steam chest. The chipping strips at the entrance of the valve ports not only facilitate the finishing up to the exact sizes, but give an increased area for steam through the valve. This is an advantage; for it is evident that the boss round the valve spindle reduces the area of these passages. The chipping pieces compensate for this reduction.

Trick Valve.—Fig. 12 shows a form of slide known as the trick valve; a design which gives a better steam opening than an ordinary valve. An inspection of the figure will render its action apparent. E is a passage in the valve which will admit steam from the left-hand side of the valve to the right-hand port, or from the right-hand side of the valve to the left-hand port, when the edge of the valve T slides past the end of the port face. The port and valve faces are so disposed that at the instant this occurs, the port will also be opened by the edge F, thus admitting steam as an ordinary slide valve. The proportions of this valve are as follows :—

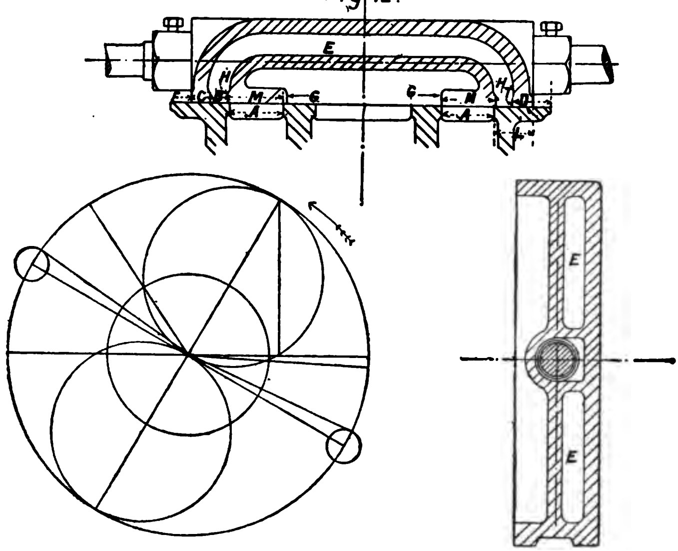
B = half required width of port opening.
A = 2 B + C, if sufficient for exhaust.
L is the outside lap, as obtained from valve diagram.
D = L.
M = half travel of valve.
Area of passage E = half area of port opening.

It is important that the edges H do not open the port before the edges G have closed; otherwise, the steam in passage E will be lost at each stroke. When the above proportions do not admit of this, B may be reduced the required amount, the reduction being

compensated by an increased opening past the edge F; which means a little extra travel. If the point of cut-off is to be maintained, the increased travel will alter the lap; and it is probable that the ports will have to be re-arranged to suit the altered circumstances.

The valve diagram calls for no comment, except that the true port opening is not the amount shown by the figure. The lead is twice the amount shown. An equalisation of release and compres-

Fig 12.



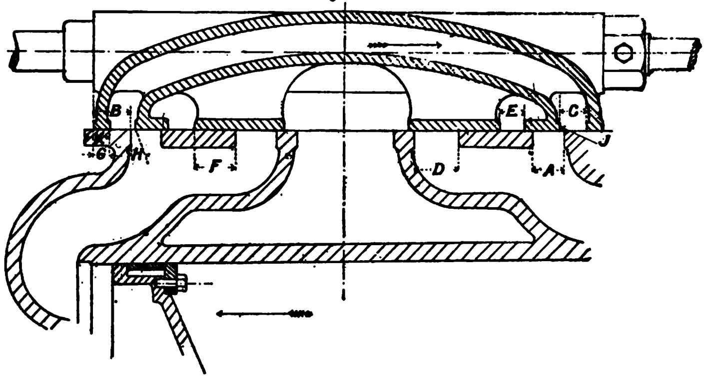
Trick Valve.

sion has been obtained by inside lap on the front end of the valve. The trick valve should have chipping strips at the edge of the ports in order to compensate for the reduction of area due to the boss of the valve spindle.

Thom's Trick Valve.—Thom's patent trick valve is an ingenious and useful invention, and has been successfully applied to many low-pressure cylinders of marine engines. In Fig. 13, the valve is shown in its central position, and the corresponding position of the piston is also indicated. Under these conditions, steam is entering from the top, through the patent passage, to the bottom side of the piston. A slight movement of the valve in the upward direction will destroy this connection; and the piston, continuing its down stroke, will compress the steam remaining in the cylinder and port at the bottom end. When it is remembered that the terminal pressure of most condensing engines is very low, the difficulty of obtaining satisfactory compression will be appreciated: and the

advantage of this valve is, that it substitutes steam of a pressure equal to that at release point, in place of a vacuum that approxi-

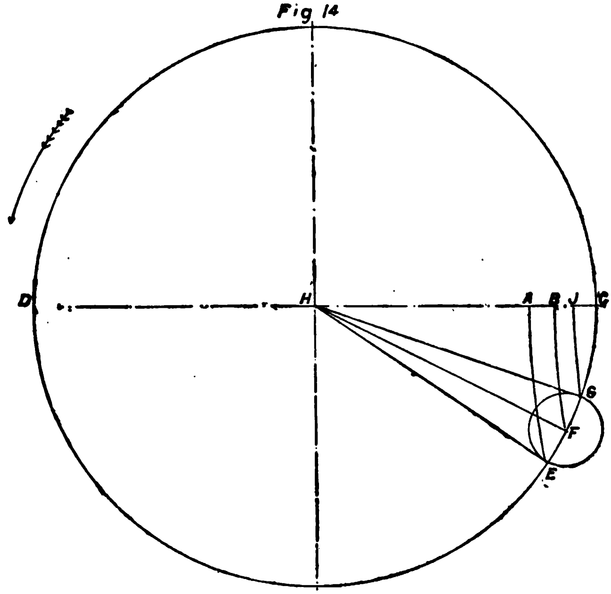
Fig 13.



Thom's Trick Valve.

mates to that of the condenser. A proper compression is thus obtained, the ports are hotter at the commencement of the next

Fig 14



stroke than they would otherwise be, and hence there is less condensation; and a saving of steam at release point pressure, and in

volume equal to the capacity of the ports and clearance, will be effected at each revolution.

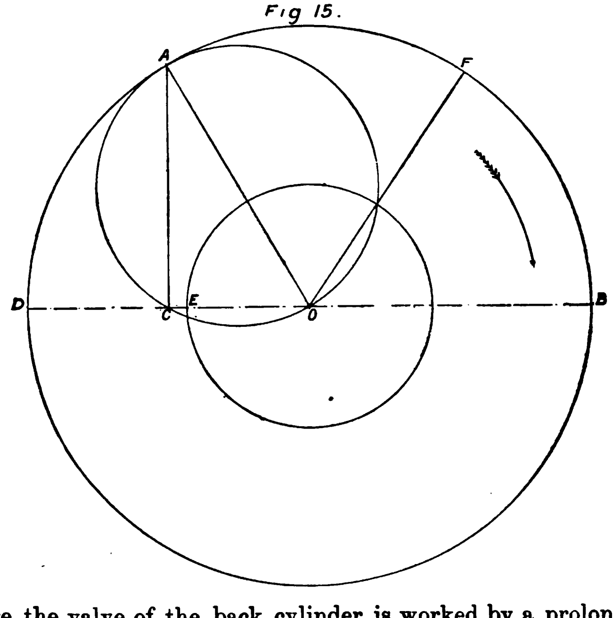
The proportions of this valve may be as follows :—

B = lap as obtained from diagram.
C = half required width of port opening (steam).
A = 2C + G.
K = B.
D + E = necessary width for exhaust.
F = half travel of valve + say $\frac{1}{2}$ inch for tightness.

It will suffice if the communication between the ends of the cylinder occur when the valve has moved in a position in which the exhaust has closed by about $\frac{3}{8}$ of an inch. The accompanying diagram will illustrate the action. Let O D represent the travel of the valve and also the stroke of the piston. Let H E be the position of the crank when compression commences, the direction of rotation being that indicated. The corresponding position of the piston is A. Make E F equal to the desired cover of the valve before communication takes place. H F is the crank position, and B the piston position when communication takes place. Let F G equal H (Fig. 13), the distance between the edge of the valve J and the edge of the port. Then communication will exist while the crank is travelling from F to G, and the piston from B to J.

Valve Diagram for Tandem Engines.—In tandem engines,

Fig 15.



where the valve of the back cylinder is worked by a prolongation of the valve spindle of the front cylinder, it is evident that the

travels and the advance angles of the two valves will be the same. Suppose the front valve already designed. Having decided upon the lead of the back valve, all necessary data are to hand for the construction of a diagram. From O as centre, draw the valve travel circle A B D. Draw O A, the given position of the eccentric in relation to the real position of the crank O B. Draw A C perpendicular to B D; and from c, mark off C E, equal to the lead. E O will be the lap. Draw the lap circle. Describe also the valve circle, giving by its intersection with the lap circle, the position of the crank at cut-off O F. E D is the maximum width of steam port opening; the area of which being known, the length is obtained. Should this be found inconveniently long, a trick valve, or a double ported valve, would perhaps set matters right in this respect; or the front valve ports might be altered so as to overcome the difficulty. O F also may be found unsuitable; in which case, a compromise between the two cylinders may be effected, in order that this objection may be removed, or at least mitigated.

Valve Spindles.—The determination of the size of valve spindles requires careful consideration, in order that they may perform their work without trembling or buckling.

Let μ = coefficient of friction between valve and port faces. (Take 0·2, which allows for weight of valve and friction of stuffing boxes.)
P = pressure of steam in valve chest, in lbs. per sq. inch.
S = working stress on valve spindle at smallest diameter. (5000 to 6000 lbs. per sq. inch for steel; 4000 to 5000 for iron.)
A = full area of back of valve in sq. inches.

Then area of spindle at smallest diameter $= \dfrac{\mu P A}{S}$; which rule agrees closely with practice. But this rule must not be applied in every case; for other considerations may modify the dimension thus found. In cases where the distance from the valve to the spindle guide is considerable, and where the eccentrics are not in line with the valve spindle, which would cause a bending action on the latter, it would be advisable to exceed the dimension found by formula.

The following table, collated from actual and recent practice, will serve to show the effect of various conditions that influence the dimensions of valve spindles :—

Description of Engine.	Boiler Pressure.	Extreme Dimensions of Valve in inches.	Distance from centre of Valve to centre of Guide in ins.	Diameter of Valve Spindle at Stuffing Box in inches.	Remarks.
Vertical triple expansion condensing, 8-in. high-press. cylinder, ...	160	$7\frac{1}{4} \times 7\frac{1}{2}$	...	$1\frac{3}{8}$	Simple slide valve. Steel spindle. No guide for spindle other than stuffing box. From centre of valve to centre of eye in valve spindle 2′ $4\frac{1}{2}$″.
13-in. intermediate cylinder for above engine, . .	...	$7\frac{7}{8} \times 10\frac{1}{2}$	...	$1\frac{3}{8}$	Same remarks apply. Maximum pressure in valve chest 75 lbs.
22-in. low-press. cylinder for above engine, . . .	...	8×18	...	$1\frac{5}{8}$	Same remarks apply. Maximum pressure in valve chest 16 lbs.
$17\frac{1}{2}$-in. locomotive, . .	140	11×14	48	$1\frac{7}{8}$	Simple slide valve. Yorkshire iron case-hardened spindle.
Horizontal twin compound condensing, 25-in. high-press. cylinder, . . .	100	20×22 main valve, $7\frac{1}{2} \times 21$ cut-off valves,	96	$2\frac{1}{2}$ main, $2\frac{1}{8}$ cut-off,	Meyer expansion gear. Steel spindles.
44-in low-press. cylinder for above engine, . . .	...	40×42 main valve, 10×42 cut-off valves,	96	$2\frac{1}{2}$ main, $2\frac{1}{8}$ cut-off,	Double-ported Meyer expansion gear. Steel spindles. Maximum pressure in valve chest 32 lbs.
60-in. low-press. cylinder of compound vertical condensing engine, ...	...	52×48	77	$3\frac{1}{8}$	Double-ported valve. Steel spindle. Maximum pressure in steam chest 25 lbs.
Horizontal tandem compound condensing, 13-in. high-press. cylinder, . . .	140	$11\frac{1}{2} \times 12$ main valve, 5×12 cut-off valves,	66	$2\frac{1}{8}$ main, $1\frac{1}{2}$ cut-off,	Meyer expansion gear. Short valve at each end of steam chest, fitted with relief ring. Main valve spindle prolonged to work low-press. valve. Steel spindles.
27-in. low-press. cylinder for above engine, . . .	...	$14 \times 22\frac{1}{4}$	60	$1\frac{3}{4}$	Simple slide valve. Steel spindle. Maximum pressure in valve chest 38 lbs.
Horizontal condensing, 27-in. cylinder, ...	60	$17\frac{1}{2} \times 21$ main valve, $5\frac{1}{2} \times 21$ cut-off valves,	40	$1\frac{3}{4}$ main, $1\frac{1}{2}$ cut-off,	Meyer expansion gear. Steel spindles.
Compound tandem vertical condensing, 20-in high-press. cylinder, . . .	80	11×20 main valve, 6×20 cut-off valves,	54	$2\frac{1}{2}$ main. $2\frac{1}{8}$ cut-off,	Meyer expansion gear. Spindles prolonged to work low-press. valves. Steel spindles.
38-in. low-press. cylinder for above engine, . . .	...	21×28 main valve, $6\frac{3}{4} \times 21$ cut-off valves,	32	2 main, $1\frac{5}{8}$ cut-off,	Meyer expansion gear. Steel spindles. Maximum pressure in valve chest 21 lbs.
Horizontal twin compound condensing, 20-in. high-press. cylinder, . . .	120	$18 \times 22\frac{1}{2}$ main valve, $13 \times 22\frac{1}{2}$ expansion valve,	48	$1\frac{7}{8}$ main, $1\frac{5}{8}$ expansion,	Hartnell's automatic expansion gear. Steel spindles.
40-in. low-press. cylinder for above engine, . . .	...	48×33 main valve, 12×33 cut-off valves,	105	$2\frac{1}{4}$ main, 2 cut-off,	Double-ported Meyer expansion gear. Steel spindles. Maximum pressure in valve chest 30 lbs.

Description of Engine.	Boiler Pressure.	Extreme Dimensions of Valve in inches.	Distance from centre of Valve to centre of Guide in ins.	Diameter of Valve Spindle at Stuffing Box in inches.	Remarks.
20-in. locomotive, . . .	170	11 × 19	53	$2\frac{1}{8}$	Simple slide valve. Steel spindle.
Horizontal twin compound condensing, 40-in. high-press. cylinder, . . .	90	37 × 35 main valve, 13 × 35 cut-off valves,	123	3 main, $2\frac{1}{2}$ cut-off,	Double-ported Meyer expansion gear. Steel spindles.
74-in. low-press. cylinder for above engine, . . .	...	54 × 62 main valve, 14 × 60 cut-off valves,	123	3 main, $2\frac{1}{2}$ cut-off,	Double-ported Meyer expansion gear. Steel spindles. Maximum pressure in steam chest 30 lbs.
10-in. vertical engine, . .	60	$6\frac{1}{4} \times 9\frac{1}{4}$	17	1	Simple slide valve. Steel spindle.
18-in. horizontal engine, .	50	$8\frac{3}{4} \times 16$	$30\frac{1}{8}$	$1\frac{1}{4}$	Simple slide valve. Wrought-iron spindle.
Horizontal tandem compound, 19-in. high-press. cylinder, .	70	$22\frac{1}{2} \times 15\frac{1}{2}$ main valve, $5\frac{1}{2} \times 15\frac{1}{2}$ cut-off valves,	72	2 main, $1\frac{1}{2}$ cut-off,	Meyer expansion gear. Steel spindles. Main valve spindle prolonged to work low-pressure valve.
28-in. low-press. cylinder for above engine, . . .	...	$19\frac{1}{8} \times 27$	$55\frac{1}{8}$	$1\frac{1}{2}$	Trick valve. Steel spindle. Maximum pressure in valve chest 20 lbs.
13-in. locomotive, . . .	120	$8\frac{1}{8} \times 11\frac{3}{4}$	48	$1\frac{1}{2}$	Simple slide valve. Yorkshire iron case-hardened spindle.
10-in. locomotive, . . .	130	$7 \times 8\frac{1}{8}$	45	$1\frac{1}{4}$	Simple slide valve. Yorkshire iron case-hardened spindle.
Horizontal twin compound condensing, 18-in. high-press. cylinder, . . .	120	$16 \times 7\frac{1}{8}$ main valve, $12 \times 17\frac{1}{8}$ expansion valve,	66	2 main, $1\frac{1}{2}$ expansion.	Hartnell's automatic expansion gear. Steel spindles.
32-in. low-press. cylinder for above engine, . . .	...	33 × 27 main valve, $8\frac{1}{4} \times 27$ cut-off valves,	87	2 main, $1\frac{3}{4}$ cut-off,	Double-ported Meyer expansion gear. Steel spindles. Maximum pressure in valve chest 34 lbs.
16-in. locomotive, . . .	120	$9\frac{1}{4} \times 15\frac{1}{2}$	46	$1\frac{1}{2}$	Simple slide valve. Steel spindle.
7-in. horizontal engine, . .	95	$3\frac{3}{4} \times 5\frac{1}{4}$	15	$\frac{3}{4}$	Simple slide valve. Steel spindle.
34-in. vertical engine, . .	65	22 × 32 main valve, 17 × 32 expansion valve,	66	$2\frac{1}{3}$ main, $1\frac{5}{8}$ cut-off,	Automatic expansion gear. Steel spindles.

Valve Pins.—In consequence of the slight movement on the pins of ordinary valve gears a large surface pressure may be allowed. 1500 lbs. per square inch of surface is frequently taken, but this figure should be used with caution, and varied according to circumstances. Where the reciprocations exceed 500 per minute this pressure would be found too great, and 1100 would agree more closely with practice. The surface being understood to denote the length of the pin multiplied by the diameter, it follows that A being the area of the valve, P the maximum pressure of steam in the valve chest, μ the coefficient of friction, and p the surface pressure allowed, the surface of the pin will be—

$$\frac{AP\mu}{p}.$$

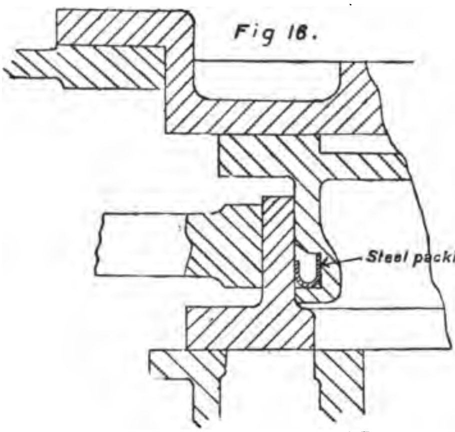
Also, if D and L equal the diameter and length of the pin, respectively,

$$D = \frac{AP\mu}{pL}\ ; \text{ and } L = \frac{AP\mu}{pD}.$$

The proportion of D to L may be as 1 : 1·25. Then,

$$D = \sqrt{\frac{\frac{AP\mu}{p}}{1{\cdot}25}}.$$

Balanced Valves.—The balanced slide valve is not regarded with much favour by many engineers, but there can be no doubt

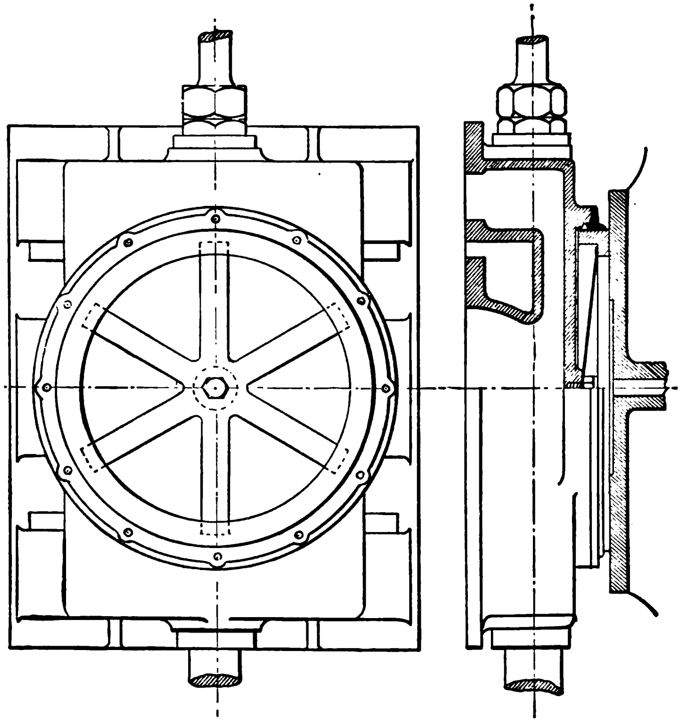


Common Form of Relief Ring.

that a well-designed and efficient arrangement is an advantage, particularly in large reversing engines operated by hand. To be a success, it is necessary that the relief rings and packings should be of the best workmanship, without which the appliance is worse than useless. Two common forms of relief rings are shown in Figs. 16 and 17.

In practice it does not appear that the presence of a balancing ring reduces the diameter of the valve spindle to any appreciable extent; it being considered unsafe to rely upon the former relieving the valve, so that in the event of the relief ring becoming ineffective the engine may still be run with safety.

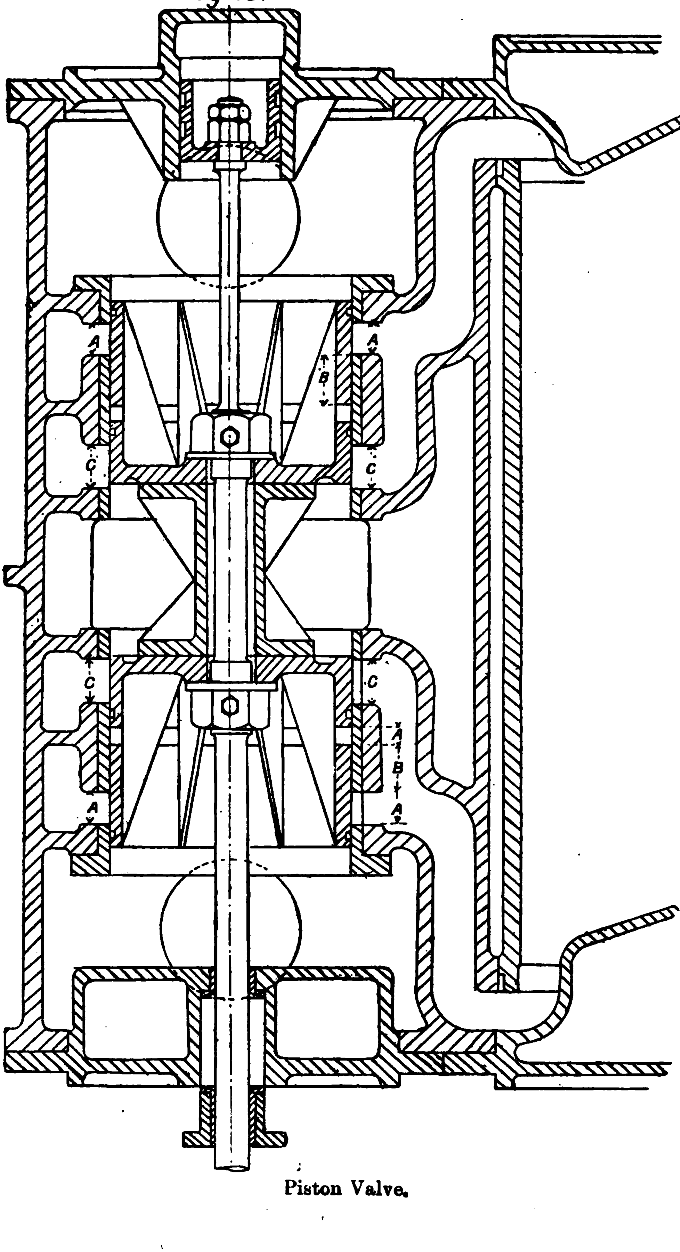
Fig. 17



Common Relief Frame.

Piston Valve.—A more effectual way of avoiding excessive friction of the valve gear is by the use of a piston valve. The first cost, however, is excessive, and its use entails large clearances. Fig. 18 is given as an example of a piston valve. The valve opens both ports for steam, but one only for exhaust. In this example packing is dispensed with. The pistons and working barrels are finished to inside and outside standard gauges. The barrels are forced into position, and secured by set screws to the main casting. When the pistons and barrels are thus carefully constructed, and have small grooves twined on the face, as shown in the figure, the escape of steam to exhaust will be so slight as to be immaterial.

Fig 18.



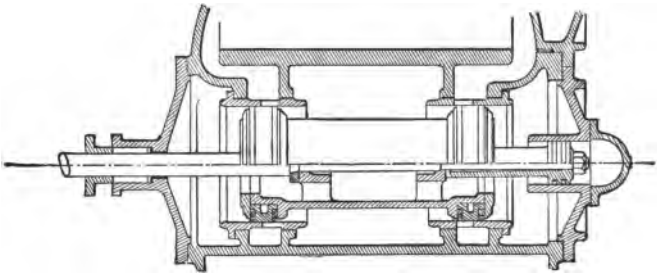
Piston Valve.

The proportions of this valve are as follows :—

A = half required width of port opening for steam.
B = half travel of valve + (say) ½ inch for tightness.
C = required width of opening for exhaust, which in this design must not be greater than the half travel of the valve.

Sometimes the steam enters between the pistons. In cases of this sort, the real position of the eccentric relative to the position

Fig 19.

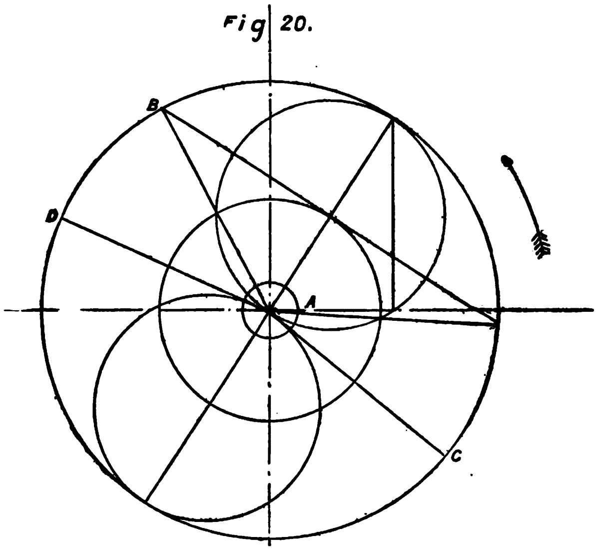


Common Piston Slide Valve.

of the crank, will be directly opposite the position indicated on the valve diagram; that is to say, the eccentric will follow, and not lead the crank.

In the above figure is shown another form of piston valve. The

Fig 20.



two pistons are part of a cylindrical casting carrying losses through

which the valve spindle passes, and to which it is secured. The packing of the pistons consists of metallic spring rings held in position by junk rings at each end of the valve. To prevent the rings springing into the ports diagonal bars are cast round the entrance of the latter. These bars, of course, contract the steam passage, and in calculating the area of steam and exhaust openings should be allowed for. The usual practice of having loose liners for the working barrels is followed; and the balance piston relieves the link motion of the weight of the valve, spindle, and other details.

The valve diagram is appended (Fig. 20), but as the construction is exactly similar to that of the slide diagram, it is unnecessary to enter upon an explanation. An inspection of the figure will show that cut-off is at A B—about 75 per cent.—and compression and release are at A C and A D respectively.

CHAPTER II.

SLIDE EXPANSION GEARS.

IT has been remarked that for an earlier cut-off than $\frac{5}{8}$ stroke, an ordinary slide valve is not suitable. In order to obtain the efficiency of steam at pressures now common, it is necessary, even in compound engines, to cut-off at a much earlier point in the stroke than can be suitably effected by a single valve; and the problem of attaining the required degree of expansion without disturbing the proper points of release and compression presents itself. The mechanism employed to attain this end is termed Expansion Gear.

Expansion Gears.—Expansion gears may be classed under three heads:—1. Expansion gears in which the cut-off is fixed. 2. Expansion gears in which the cut-off is varied by hand. 3. Expansion gears in which the cut-off is automatically controlled by the engine through the operation of a governor.

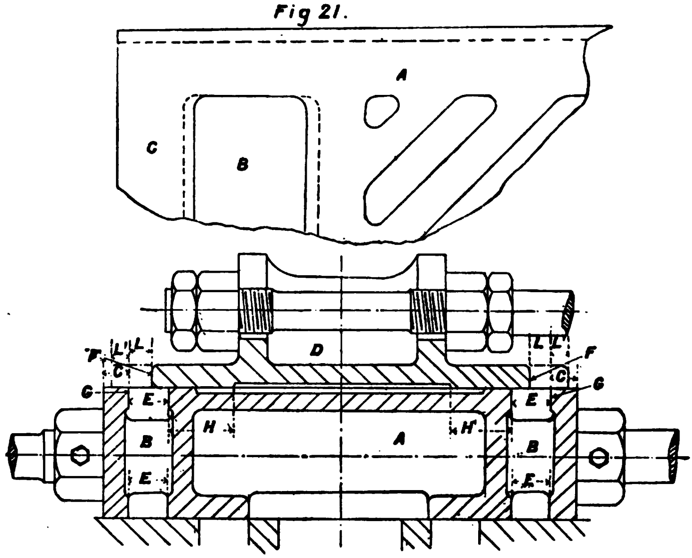
Fixed expansion gears are now little used, and are only suitable for engines having a definite and unvarying load, which condition is very rarely the case. However, as the consideration of this class embodies the principle of most expansion gears, it will be convenient to investigate it in the first instance.

In Fig. 21, A is a slide valve having steam ports B B formed at each end. This valve is virtually an ordinary slide, with the addition of the ribs C C, which form the outer side of the steam ports B B. Sliding upon the back of this valve is the expansion valve D, driven by a separate eccentric, the sole function of which is to control the admission of steam. It is easy to see that by arranging the eccentrics in a proper manner, the expansion valve can be made to slide past the outer edges G G of the ports B B, at any desired point in the stroke.

Diagram for Expansion Valves.—Before considering the expansion valve, the main valve must be designed. This is done

exactly as in the case of an ordinary slide valve. The width E equals that of the maximum opening to steam, as found by calculation. The width of the bar C is empirical.

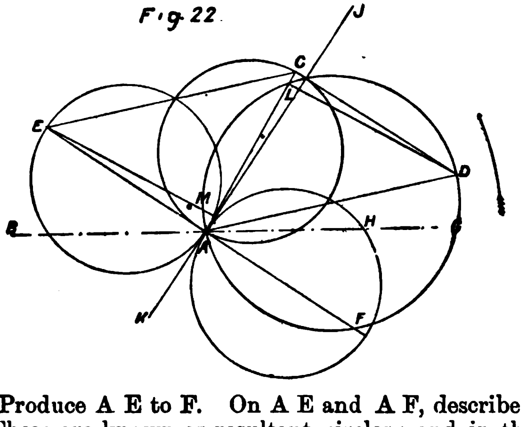
Fig 21.



Slide Expansion Valve.

In Fig. 22 let A B represent the position of the crank, and A C and A D the positions and throws of the main and expansion eccentrics respectively. Join C D, and draw E C equal and parallel

Fig. 22



to A D. Produce A E to F. On A E and A F, describe circles as shown. These are known as resultant circles; and in the manner in which the primary and secondary valve circles illustrate the

action of the simple valve, these resultant circles give the motion of the expansion valve relative to the main; and this relative motion is such as would be obtained by an eccentric whose throw and position are given by the line A E, when the main valve is supposed to be fixed, and in its central position.

In this instance the peculiarity of the Zeuner diagram is still present. The real position of the crank relative to the eccentrics, A C and A D, is A B; but in tracing the action of the steam, the motion is supposed to be in the direction of the arrow.

In each diagram, unless there is a direct statement to the contrary, the arrow shows the *assumed*, and not the real direction of rotation.

The motion will now be traced throughout a complete revolution.

Starting at the beginning of the forward stroke, the assumed crank position being A G, the expansion valve will be distant A H to the left of its central position on the main valve; which distance is given by the intersection of the crank position and the secondary resultant circle. The expansion valve continues to move from left to right until A J is reached; at which point it arrives at its central position relative to the main valve; that is to say, the centres of the two valves will coincide, and will have, relative to one another, the position shown in Fig. 21; but Fig. 21 does not show their true position relative to the cylinder ports. A J is at right angles to A F. The expansion valve now travels to the right of its relative central position until A E is reached, at which point it attains the end of its relative travel to the right. At A B, the expansion valve is still to the right, as shown by the intersection of the crank with the primary circle. At A K, relative mid position is again reached; and when the crank is at A F, the valves are at their maximum distance apart. This displacement now decreases, and at A G the revolution is completed.

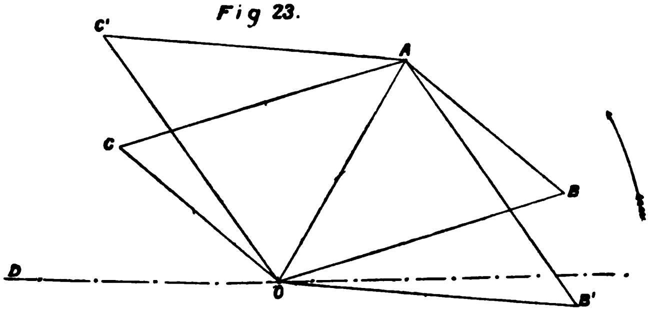
In order to show that the resultant circles give exactly the action of the two valves relative to one another, draw on A C and A D the main and expansion valve circles. Take any position of the crank, say A C. The intersection of the crank line with the main eccentric circle A C shows the distance of the main valve to the left of its central position on the ports, as explained above; and the intersection of the crank line with the expansion eccentric circle also shows the distance of the expansion valve to the left of its central position. The distance apart of the two valves relative to each other is obviously C L. The crank line A C intersects the resultant circle in M, also showing the distance apart of the two valves. It remains to show that A M = C L. Join E M and D L. Then, because E A was made parallel and equal to C D, the angle E A C is equal to the angle A C D; and the angles E M A and D L C are right angles and equal to each other (Euclid, Bk. iii., Prop. 31). Then, in the triangles E M A and D C L, because the side E A = the side C D (by construction) and the angle E M A = the angle C L D, and the

angle E A M = angle L C D, the side A M = the side C L (Prop. 26, Bk. i., Euclid's Elements).

By reasoning similar to the above, the resultant circles can be shown to give exactly the relative positions of valves for all positions of the crank, and for any form of expansion gear operated by eccentrics in the ordinary manner.

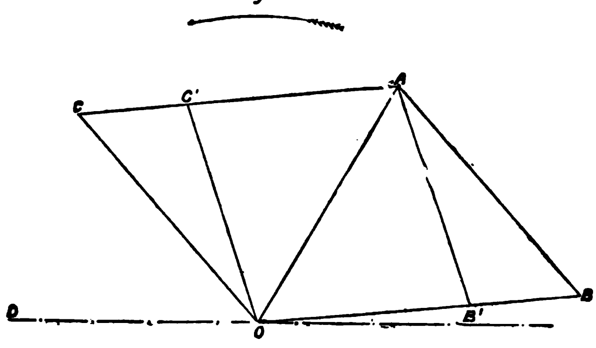
Now, consider the effect of altering the throw and position of the expansion eccentric. In Fig. 23 let O A and O B represent the

Fig 23.



main and expansion eccentrics, and O D the real position of the crank. Then O C will be the line giving the resultant circles. Let the expansion eccentric be advanced to O B′. Then O C′ is the centre on which the resultant circles are described. The expansion valve will therefore attain its central position on the

Fig 24.

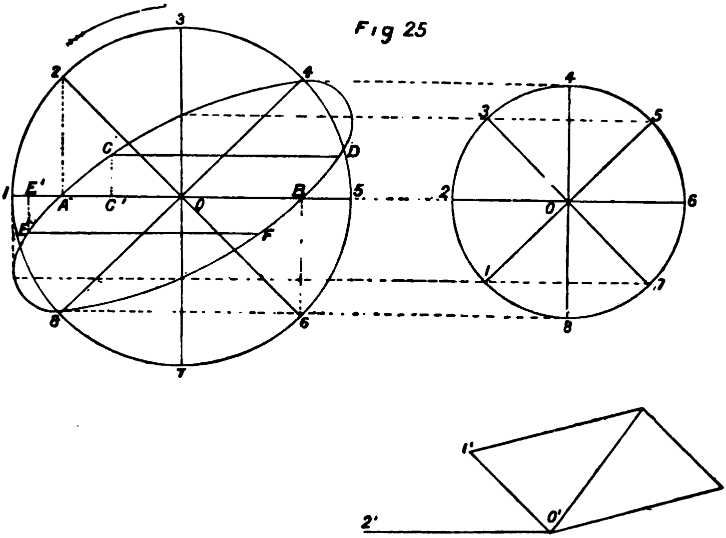


main valve earlier in the stroke, and the relative travel of the valves will be greater.

In Fig. 24 O D is the real position of the crank; O A is the main, and O B the expansion eccentric. O C is the centre of the resultant circle. Let the travel of O B be reduced to O B′. Then O C′ will be the altered position of O C. The expansion valve will therefore attain its central position on the main valve earlier in the stroke, but the relative travel will be less.

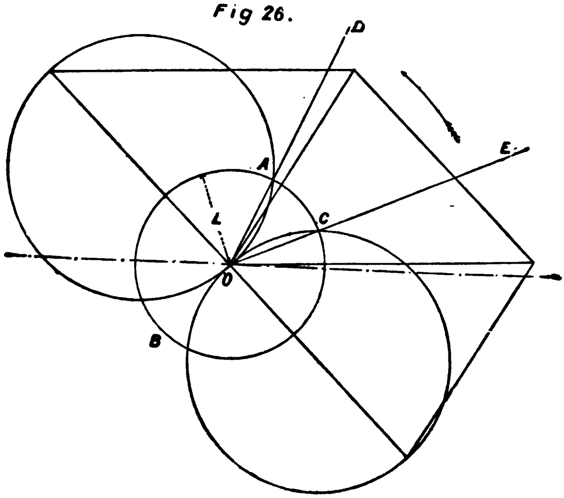
Application of Ellipse Diagram to Expansion Gears.— The resultant circles can be reduced to an ellipse. It will be seen

Fig 25



by an inspection of Fig. 25, that the angle of the centre line of the resultant circles is put back 90° from its real position in a similar

Fig 26.



manner to the advance angle in the simple valve ellipse diagram. In the present figure, A and B are shown to be central positions; 4 and 8 extreme positions.

The effect of lap will now be considered. In Fig. 26, let L equal the dimension L in Fig. 21. From O, draw O A D. This line will represent the position of the crank when the expansion valve has closed the port in the main valve. Again, suppose L is positive. The expansion valve would then have the form shown by the dotted lines in Fig. 15; and L′ (Fig. 21) would equal L (Fig. 26). Where the circle A B C cuts the secondary resultant circle, draw the line O C E. This line is the position of the crank when cut-off takes place, the lap being positive. The following statements are deduced from the foregoing observations:—When the lap is positive, the point of cut-off is given by the intersection of the lap circle with the secondary resultant circle. When the lap is negative, the point of cut-off is given by the intersection of the lap circle with the primary resultant circle. When there is no lap the point of cut-off is given by a line at right angles to the centre line of the resultant circles.

In the valve ellipse diagram, if the lap be negative, draw C D parallel to, and remote a distance equal to the lap, above the line 1 5. The line C D crosses the ellipse at C. Then C′ is the position of the piston at cut-off; 1 5 representing the whole stroke. Let the lap be positive. E F is then the line that determines the point of cut-off, it being drawn below the line 1 5. E′ denotes the piston position at cut-off.

Having premised these introductory remarks, the actual design of the valves will now be considered.

Valve Diagram.—On the diagram of the main valve draw the crank position when it is desired that the expansion valve shall cut-off. In Fig. 27, O A is the line. The position and throw of the expansion eccentric are not determined by any exact construction. The travel is usually made greater than that of the main valve. $T \times 1{\cdot}2$ is good practice; T being the travel of the main valve. As to the angle of advance, it has been shown that increasing this, increases the diameter of the resultant circles; so that other things being equal, the cut-off will be quicker Hence, in practice, it is found that the angular advance is considerable. Eighty degrees in advance of the main eccentric is very suitable, but some makers have a constant angle, which in several cases is 90°; that is, directly opposite the crank.

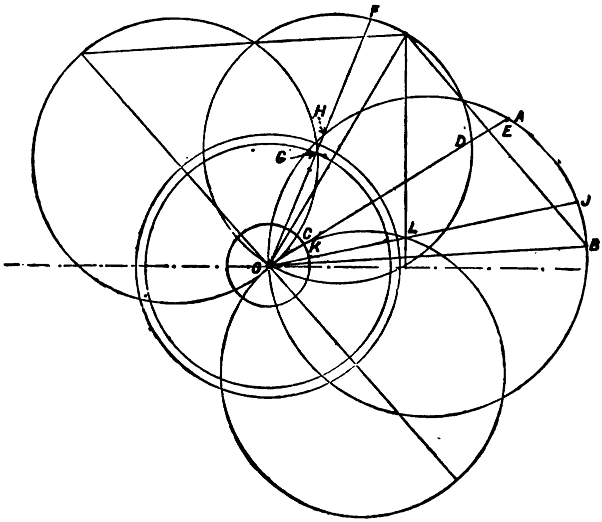
Let O B (Fig. 27) represent the throw and position of the expansion valve eccentric decided upon. Draw the resultant circles. The secondary circle intersects O A at C. O C is the amount of positive lap on the expansion valve.

When the cut-off is nearly at right angles to the centre line of the resultant circles, it becomes difficult to see the exact point at which intersection takes place. If, however, a circle be described on the expansion eccentric line O B, the difficulty can be avoided; for the distance O C is also given by the distance between the intersection of the main valve circle with the cut-off line, and the intersection of the expansion eccentric with the same line. In the

figure, D and E are the points of intersection; and D E equals O C. Take another position, say O F. O G is the necessary lap, which is also given by H F. Here the lap is negative.

With regard to the port opening for any particular crank position, it is evident that when the lap is positive the port opening is given by the distance between the expansion valve lap circle, and the intersection of the resultant circle with the crank line. Thus, in crank position O J, the port opening is K L. But the actual port opening can never be greater than the width of the port

Fig 27.



through the main valve; so that it would be more correct to say that K L is the distance of the edge F (Fig. 21) from the edge of the main valve port B. Suppose the lap O C is negative. Then the distance of edge B from edge G, would be L O + O K, for the crank position O J.

Design of Expansion Valves.—Referring now to Fig. 21, it will be noticed that the expansion valve is not machined the whole length of the face, but only for a width, H, past the edge of the port in the main valve. In order that steam shall not enter the port B from the inner edge of the face of the expansion valve, the distance H should be at least equal to the diameter of the resultant circles, which give the amount of relative motion of the two valves.

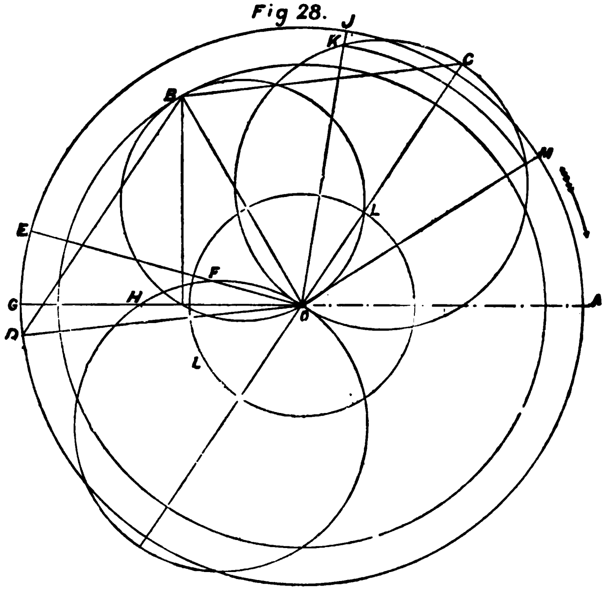
In expansion gears of this class, the cut-off should be equalised. The required adjustment cannot be found by Zeuner's diagram, but may be obtained by plotting down the crank and eccentric circles, and moving tracings of the valves into position for the front and back strokes. In practice this is seldom done. The valve spindle

is screwed past the nuts on each side, as shown in the figure, so that the cut-off may be adjusted when the engine is erected.

Meyer Expansion Gear.—Of variable hand expansion gears, the well-known Meyer gear is by far the most common. In this arrangement, a range of expansion is obtained by varying the lap of the cut-off valves.

In Fig 28, O A is the real position of the crank relative to the main eccentric O B. L L is the outside lap circle of the main valve, and cut-off occurs at O C. Decide upon the travel of the expansion valve, and from O as centre, and with radius equal to the throw of the eccentric, describe the circle A E D. Draw B D parallel to O C; and join O and D. Then O D is the position of

Fig 28.



the cut-off eccentric relative to the position of the crank O A. By this construction the centre line of the primary resultant circle will coincide with the line of the main valve cut-off. The position O D is not imperative; but for reasons which will be made known shortly, it is a most convenient setting.

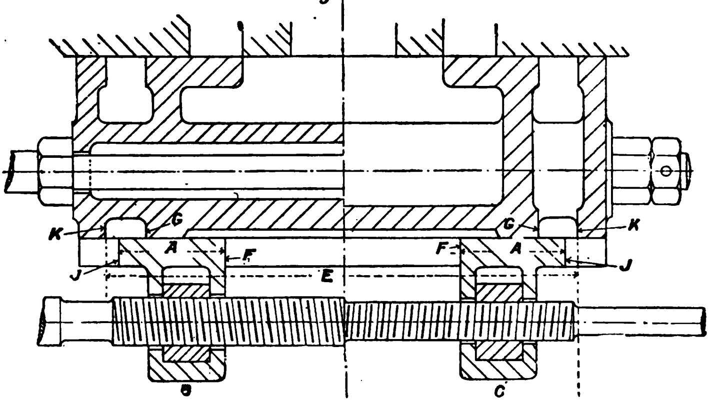
Now, the nut of the expansion plate B (Fig. 29) being screwed by a right-handed thread, and the nut of plate C being screwed by a left-handed thread, the plates can be made to approach to or recede from one another, according to the direction of rotation of the valve spindle. This movement of the plates will alter the point of cut-off to any desirable extent.

Let the required range of expansion be from the beginning of the stroke, to the point of cut-off by the main valve.

To find the position of the expansion plates relative to the main valve, it is only necessary to draw the crank line for the required point of cut-off; for its intersection with either of the resultant circles determines the position. Take the position of the crank O E (Fig. 28). The line O E cuts the secondary resultant circle at F. O F is the required positive lap; or the amount that the edges J, of the plates B and C, overlap the edges K of the main valve, when B and C are central on the main valve, which position is shown in Fig. 29.

Take the point of earliest cut-off O G (Fig. 28). Here O H is the required positive lap. At the latest cut-off O C, the lap will be O C, and as O C intersects the primary resultant circle, it will be negative. It follows, therefore, that O H + O C is the movement

Fig 29.



Meyer Expansion Gear.

of each plate on the spindle, in order to vary the cut-off from O G to O C.

The reason for choosing O C for the centre line of the resultant circles will now be stated. At crank position O C the expansion plate will be at its extreme position to the right, relative to the main valve. But the negative lap necessary for this grade of expansion is equal to O C, so that it is evident the plate will re-open the port immediately after O C. The main valve, however, has cut-off at the same instant; therefore, re-opening by the expansion valve will not affect the efficiency of the gear. The latest possible cut-off by the expansion valve is given by the centre line of the primary resultant circle. When this centre line falls on, or after, the line of main valve cut-off, re-opening, as previously stated, is immaterial. For example, take a crank position O J. O K will be the negative lap. The plate attains its extreme relative position at O C. It will now recede and re-open the port when O M is

reached, but as cut-off by the main valve has already occurred, this re-opening is of no consequence.

Application of the Diagram.—There is another way in which the port in the main valve may be re-opened, and that is by the edges F of the cut-off plates sliding past the edges G of the main valve.

Let P = diameter of resultant circles.
W = width of port in main valve (E in Fig. 29).
L = lap of expansion valve for earliest cut-off.

Then, in order to avoid this re-opening, the width A (Fig. 29) should be

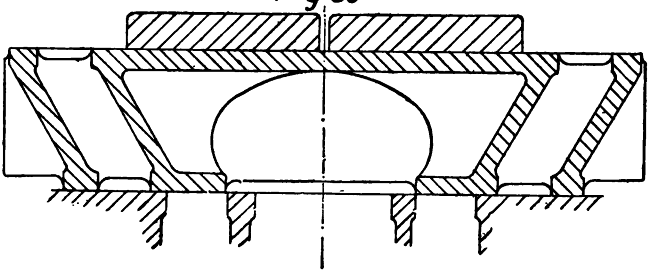
$$P + W \pm L.$$

For positive lap take + L; and for negative lap − L.

It is necessary to add a little extra width to the dimension given by the foregoing expression to ensure tightness. A quarter of an inch will, in most cases, be sufficient addition.

The diagram shows that O C (Fig. 28) is the greatest negative lap. The dimension E in Fig. 29 must be great enough to allow of both plates being set to this negative lap. Therefore, E must not be less than 2 A + twice the greatest negative lap. In some cases it may be necessary to slope the ports in order to obtain E. Fig. 30 illustrates a case in point.

Fig 30



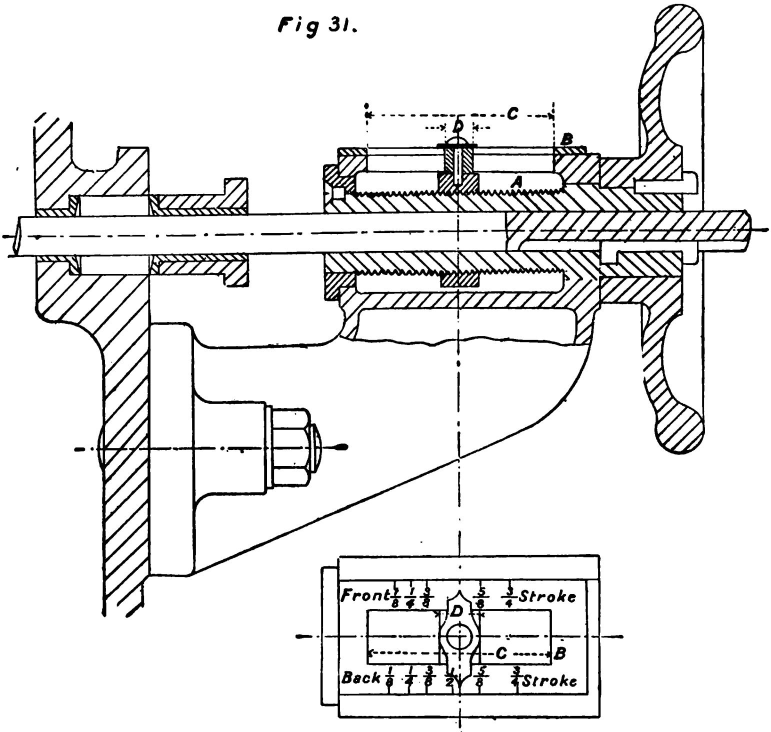
Expansion Regulator.—In order that the spindle of the cut-off valves may be revolved to vary the expansion, it is prolonged through a stuffing box at the back of the valve chest; and has a wheel at or near the end of it. An index is usually provided, to indicate the grade of expansion. Fig. 31 shows a neat arrangement of cut-off regulator. In this design, the wheel is keyed on the brass sleeve A, the latter being mounted on the valve spindle by means of a feather key, thus permitting of a horizontal movement of the valve spindle, whilst the regulating wheel is at rest. The brass plate B is stamped, so that the finger correctly indicates the grade of expansion. The minimum length of the slot C in the regulator is given by the expression

$$T + \frac{M N}{n}.$$

Where T = thickness of finger nut (D in Fig. 31).
M = movement of each plate in varying the cut-off from earliest to latest.
N = number of threads per inch in nut of cut-off valves.
n = number of threads per inch on sleeve of regulator.

The joint of the expansion-valve spindle with the eccentric-rod must be of such design as will allow the spindle to turn freely round.

Fig 31.

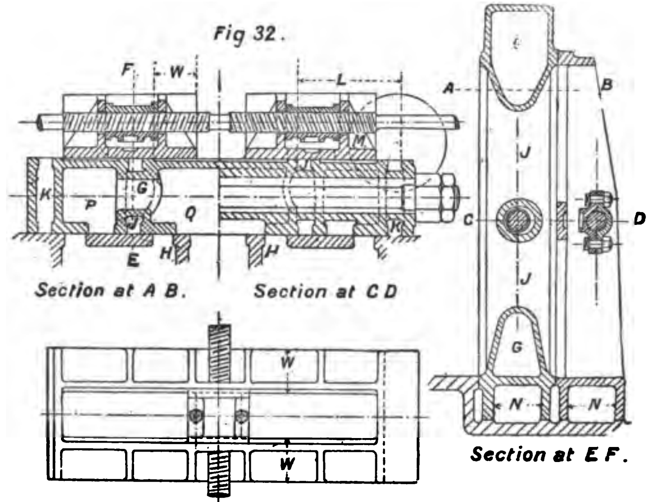


Expansion Regulator.

Double-Ported Meyer Gear.—The Meyer gear is shown in its double ported form in Fig. 32. The passages G G conduct the exhaust from P to Q. The sum of their areas should equal that of the steam port H. The ports J J are shown widened out to compensate for the reduction of area caused by G G. The area of ports J J and K K, should in all cases be equal to the area of the steam ports. The dimension L requires attention, or it may happen that when the main valve is open for lead, the plate M will cover the port J, and steam will not be admitted at the proper time. In order that such a thing may be avoided, L should not be less than

$$l + W + N + P.$$

Where l = lead of main valve at each port.
W = width of cut-off plate.
N = lap at latest cut-off, which will, in most cases, be negative.
P = distance of cut-off valves from relative central position, when crank is on dead centre.



Double-ported Meyer Gear.

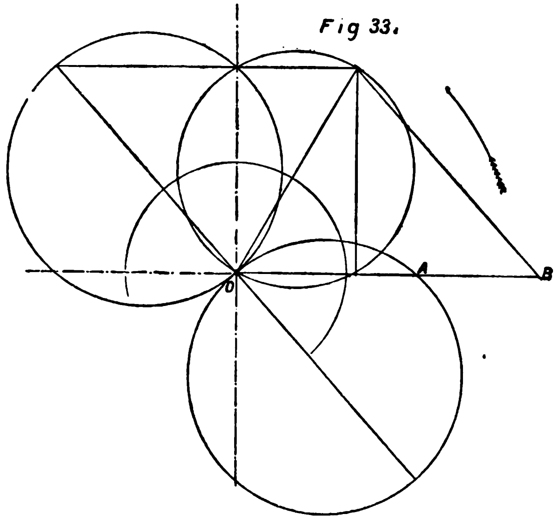
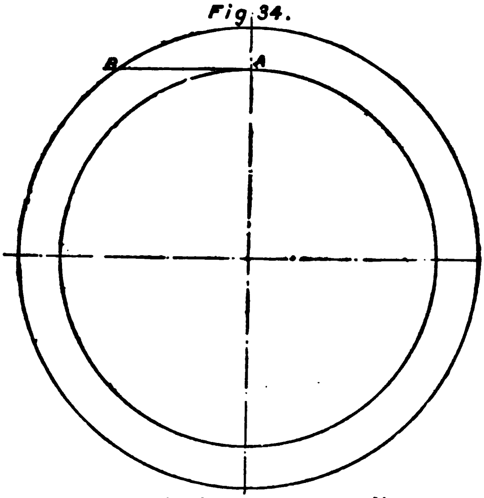


Fig. 33 is the diagram for the gear, and O A = P in the foregoing expression, and is the distance of the cut-off valves to the left of the relative central position when the crank is at O B. The expansion

valve will open the port J at a quicker rate than the main valve will open to steam; which is as it should be. In every case, it is advisable to make L greater than the dimension given by the above expression.

The nuts of the expansion valves are shown divided. This permits of the valve spindle being kept the same diameter for each nut. The recess at the back of each nut is provided for the reception of a spring, which tends to keep the plate on the face of the main valve. In order that the spindle may be passed through the valve chest when the expansion plates are in position, it is necessary to have sufficient clearance to press back the back half of each nut, to clear the thread of the valve spindle. In Fig. 34, the inner circle represents the diameter of the valve spindle at the bottom of the thread, and the outer circle the diameter at the top of the thread.

Fig. 34.



It will be necessary to push the nut back a distance equal to A B, before the spindle can be passed into position. When the steam chest is big enough to allow of the expansion plates being placed with their faces to the main valve spindle, the cut-off spindle can be passed, and the plates afterwards brought into their proper position. Then the above condition is not imperative.

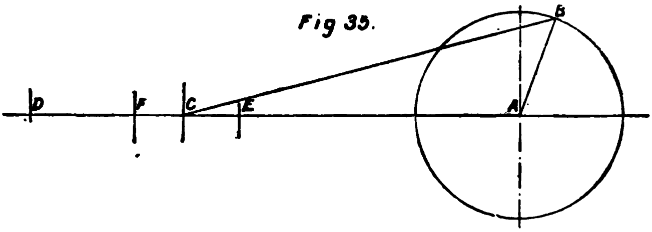
The circle to the right of the plan view in Fig. 32 represents the steam inlet pipe. The main valve is, of necessity, of large size, and it was desirable to keep the steam chest as small as possible. Both the valves slide over the supply pipe, but in consequence of the deep feet N, the orifice is not obstructed to any serious extent.

With Meyer gears, when the cut-off is equalised for any one grade of expansion, it will not be equal for the other grades. In practice, it is usual to equalise the particular grade of expansion which corresponds to the normal load on the engine.

Reynold's Valve Diagram.—The action of the Meyer gear is very clearly shown by means of the Reynold valve diagram. It will be convenient to consider the diagram for a simple valve before treating of expansion gears.

In Fig. 35, A B represents the crank, and B C the connecting-rod of an engine. The piston-rod is suppressed; and therefore D and E are the extreme positions of the piston, and F is the central position. Divide the crank-pin circle into any number of equal parts; 36 divisions is the most suitable number. The corresponding piston positions for each of the crank-pin positions can be readily obtained. Thus, taking crank position A B, F C is the displacement of the piston from its central position.

Fig 35.



Reynold's Valve Diagram.

Now, consider the line A B, in Fig. 36, to be the crank-pin circle of Fig. 29, drawn out into a straight line. Let this line be divided into 36 equal parts. Then each division will correspond to an angle of 10 degrees on the crank-pin circle. The distance of the piston from its central position, for each of these 10 degrees is obtained from Fig. 35, and transferred to Fig. 36; and a curve similar to H K J is obtained. The total breadth of the curve will be the stroke of the piston to the scale chosen. The piston is at its extreme position at H. At D, where the curve crosses the axis A B, the piston is in its central position, and the crank at 87 degrees from the starting point. K denotes that the end of the front stroke is attained. From K to J is occupied by the back stroke.

Reynold's Valve Diagram for Single Valve.—A similar curve must now be obtained for the valve. The extreme horizontal breadth of this curve will represent the travel of the valve. Any convenient scale for the breadth may be chosen, but the vertical axis of the curve must be the same length as the piston curve. Let the angular advance of the eccentric be 40 degrees. O on A B is the position for crank dead centre; so that if the valve curve cross the vertical 40 degrees before O, the curves will be in their correct position.

Take any position, say L L. The piston is distant N O from its central position; and at the same time, N P is the displacement of the valve from its central position. At M M the valve is at mid

position on the ports, and the intersection of M M with the piston curve indicates the then position of the piston. Let the outside lap be known. Draw C C parallel to A B, and removed a distance equal to the known outside lap from A B. Admission occurs at C. Draw C Q horizontally. Its intersection with the piston curve gives the piston position. This position may be projected vertically to a scale of percentages, and the position at once read off.

E E is a line parallel to A B, and G is the inside lap. Release is at E, and compression at E′. Let it be required that release shall occur at 93 per cent. in both front and back strokes. From 93 per cent. on the scale for the forward stroke, draw R S vertically, to intersect the piston curve at S. From S draw S F horizontally. Then, if the inside lap on the front end of the valve be equal to the distance between F and the vertical A B, release will occur at 93 per cent. From 93 per cent. on the scale for the back stroke, draw U V vertically, to intersect the piston curve at V. Project V horizontally, to intersect the valve curve at W. Then W X is the amount of inside lap necessary to release at 93 per cent. in the back stroke. If the intersection of these horizontals with the valve curve fell on the axis A B, no inside lap would be necessary; but if the intersections fell on the opposite side to E E′, negative inside lap would be required.

Reynold's Valve Diagram for Meyer Gear.—Take now the case of an expansion gear. Having, in the manner previously explained, obtained a curve for the expansion valve, it remains to connect this curve with the piston and valve curves in such a manner as will give the true positions of the piston, main valve, and expansion valve for any point in the stroke. It will be found convenient to plot down the main valve and piston curves on one sheet, and the expansion valve curve on a separate sheet of tracing paper, so that the latter may be set to any desired position, and the modifications effected by altering the angle of advance observed.

Fig. 37 shows the piston, main, and expansion-valve curves in position. The angular advance of the main eccentric is the same as in the previous case—namely, 40 degrees. The expansion-valve curve is placed for an angular advance of 90 degrees—that is, directly opposite the crank. A horizontal line drawn at any point of the crank-pin line A B, will give the then positions of piston and valves. And conversely, if any piston position be chosen, a vertical drawn from that position will intersect the piston curve, and a horizontal drawn from the intersection will give the valve positions.

Take the crank position C D. E F is the distance of the piston from its central position, and E H and E G will be the respective displacements of the main and expansion valves from their mid position.

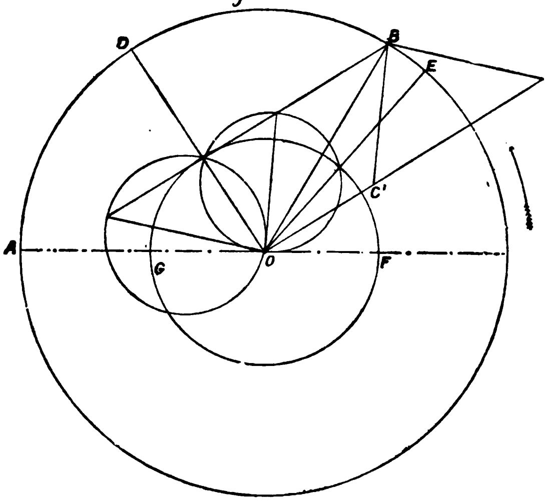
Let it be required to cut off at 20 per cent. of the stroke. From 20 per cent. on the scale for the front stroke, draw the vertical to intersect the piston curve at J. From J draw the horizontal,

crossing the main and expansion curves at K and L respectively. K L is the necessary negative lap to accomplish this grade of expansion in the front stroke. From 20 per cent. on the scale for the back stroke, draw the line vertically, to meet the piston curve at M. From M, draw the horizontal, intersecting the valve curves at O and N. O N is the negative lap for the front end of the valve, to effect this grade of expansion. If the horizontals K J and M O crossed the curves at P and Q—that is, where P and Q cross each other—there would be no lap whatever; but if the verticals crossed on the opposite side of P and Q, the lap would be a positive quantity.

The equalisation of cut-off for any particular grade of expansion having been effected, it is easy to observe the inequality for the other grades; and herein lies the chief advantage of the diagram. The expansion-valve curve being drawn on a separate sheet of tracing paper, the angular advance may be altered by merely moving the position of the tracing; the effect of which is instantly manifested. This is another merit of the Reynold's diagram.

Classes of Automatic Expansion Gear.—Automatic expansion gears are made to vary the cut-off:—1. By varying the travel

Fig 38.



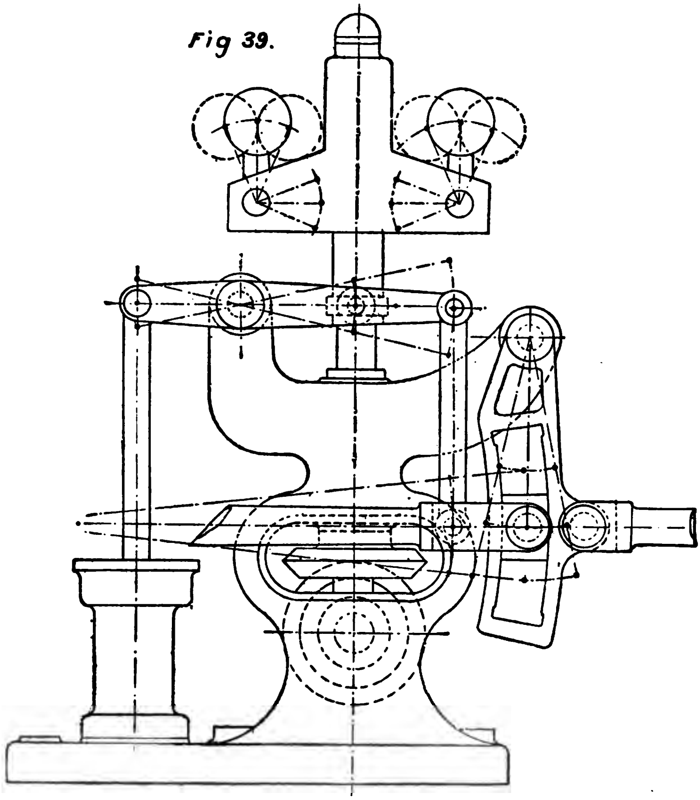
of the expansion valve. 2. By varying the angular advance of the expansion eccentric. 3. By varying both angle and travel of the main eccentric.

In Fig. 38, O A is the crank, O B the main eccentric, and O C the expansion eccentric. Let the circle F G be drawn with radius equal to the negative lap. Cut-off will occur at crank position O D. Now, suppose the throw of O C to be reduced to O C′. Cut-off then

occurs at O E. This diagram illustrates the principle of expansion gears of the first class.

Hartnell's Governor.—Fig. 39 is a sketch of the well-known Hartnell governor combined with a link expansion gear. When the engine is running at the normal speed, the die in the swinging link occupies the position indicated. When an increase of speed takes place, the balls fly out, raising the radius rod, which shortens the travel of the expansion valve. Cut-off will therefore be earlier in the stroke. When the speed falls below the normal, the radius

Fig 39.



Hartnell's Governor.

rod is lowered. This action increases the valve's travel, and cut-off is later. The governor itself is a spring-loaded one. Between certain limits, any desired speed may be maintained by adjusting the helical spring in the head.

Diagram for Hartnell's Expansion Gear.—No exact construction can be given to determine all particulars of this gear. Generally, the positions of the crank at early and late cut-off, and the maximum and minimum travels of the expansion valve would

be decided upon. But it is necessary to assume either the angular advance or the lap of the expansion valve before a diagram can be constructed.

Let O E, Fig. 40, be the line of latest cut-off in relation to the crank position O N revolving as shown; O A the throw and position of the main eccentric, and O B and O B′ the throws and position of the expansion eccentric for maximum and minimum travels respectively. O C is the resultant circle corresponding to O B; and O D the resultant circle for O B′. O E intersects the resultant circle for maximum travel, at F. O F is the required negative lap. Draw the negative lap circle F G, intersecting the resultant circle for minimum travel at G. O H is the position of the crank at earliest cut-off. It may be that this range of expansion

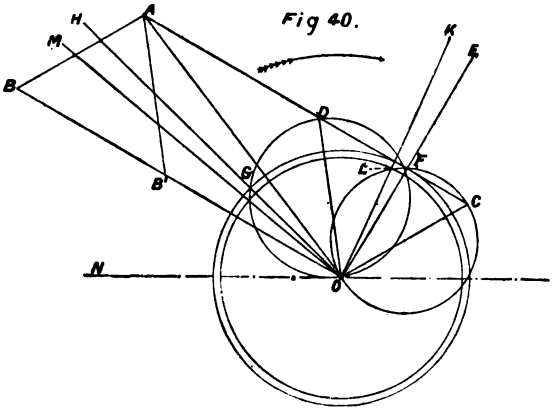


Diagram for Hartnell's Expansion Gear.

is not sufficient. Decreasing the negative lap would not alter the total range of expansion, as far as the cut-off valve is concerned, but would cause O E and O H to fall earlier in the stroke. But O K is the cut-off by the main valve. O E can be made to coincide with O K, by giving negative lap O L. This will alter O H to O M; which alteration has increased the actual range of expansion by the angle H O M. Decreasing the angular advance of the expansion eccentric would also increase the range of expansion, as may be shown by construction.

Application of the Diagram.—The valves for the Hartnell gear may assume the form shown by Fig. 41, the proportions being as follows :—

The sum of the width of the ports C, may be equal to one and a-half times the width of port E, to ensure a good steam line.

$$A = D + C - L, + \text{say } \tfrac{1}{4}\text{-inch for tightness.}$$

Where D = diameter of largest resultant circle.
C = width of port (see Fig. 41)
L = negative lap.

Note. — $-L$ would become $+L$ if lap were positive.

B should be not less than $\frac{L}{N} + D$.

Where L = lead of main valve.
N = number of ports on each side of expansion valve.
D = greatest distance of expansion valve from its relative central position, when crank is on dead centre.

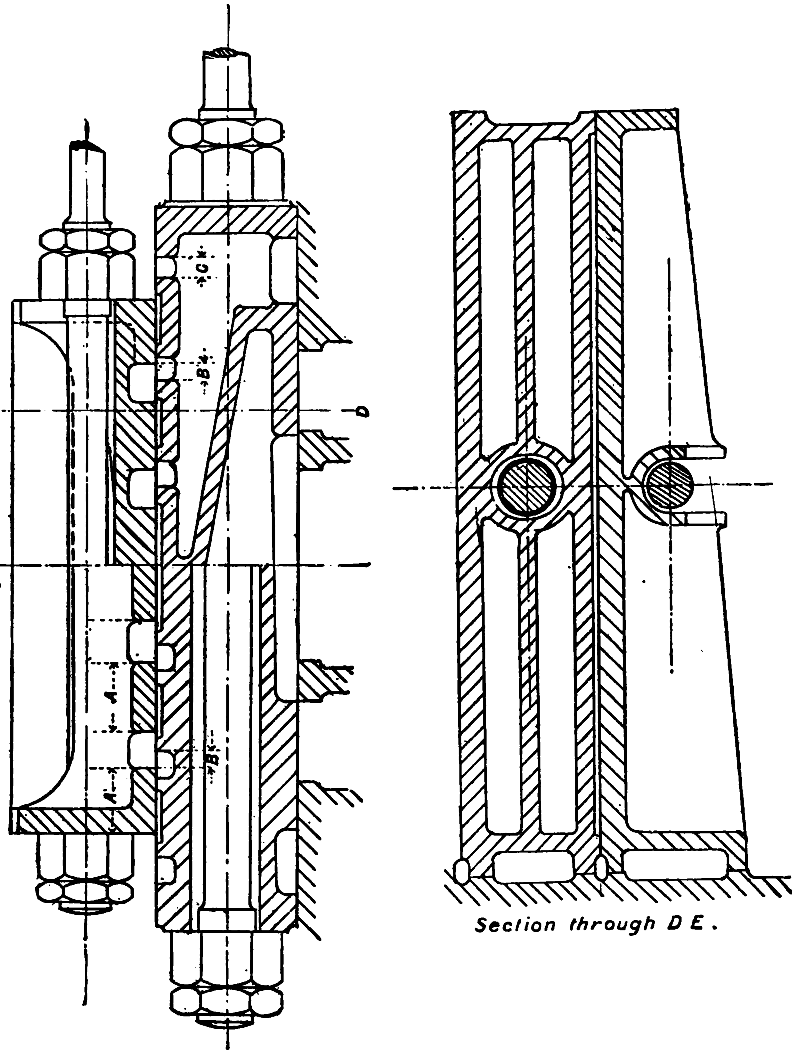
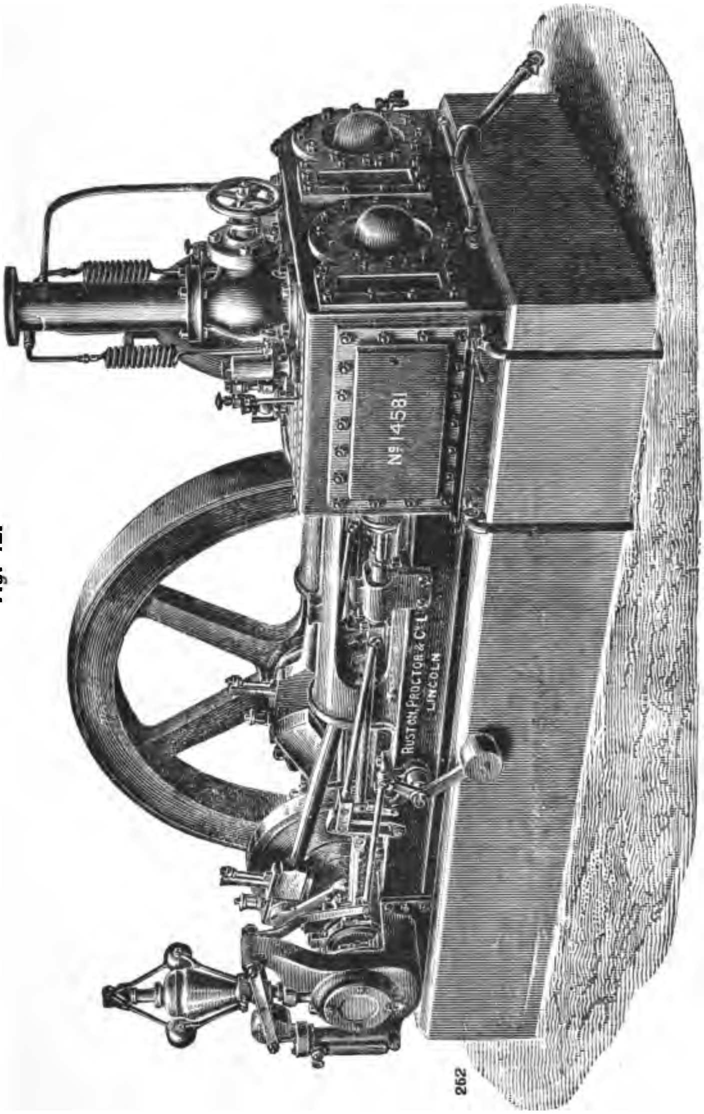


Fig. 41.

Section through D E.

In practice, B would be found to exceed the dimension given by the above expression. The ports are often divided equally along the face of the valve. The rules just given can then be employed to ascertain whether the action is satisfactory or not.

Fig. 42.

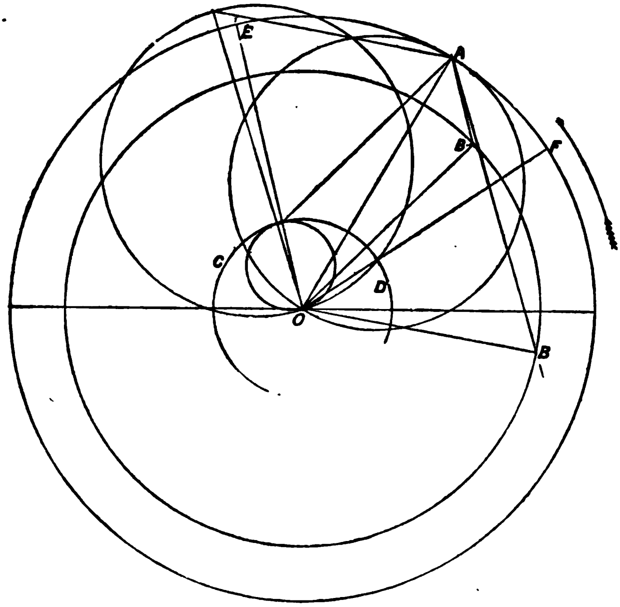


Automatic Expansion Gear.

Ruston-Proctor's Expansion Gear.—Fig. 42 illustrates the automatic expansion gear made by Messrs. Ruston, Proctor & Co., Lincoln. The governor is of the Porter type; and is actuated from the crank shaft by means of bevel gear. The radius link is pivoted on the bottom end, and the lifting link from the governor raises or lowers the radius rod as circumstances require. A dash-pot is fitted to give steadiness to the governor, and the balance weight relieves the latter from all encumbrance due to the radius rod.

Crank-Shaft Governor Expansion Gear.—It has been remarked that the grade of expansion can be varied by altering the angular advance of the expansion eccentric. Fig. 43 is a diagram

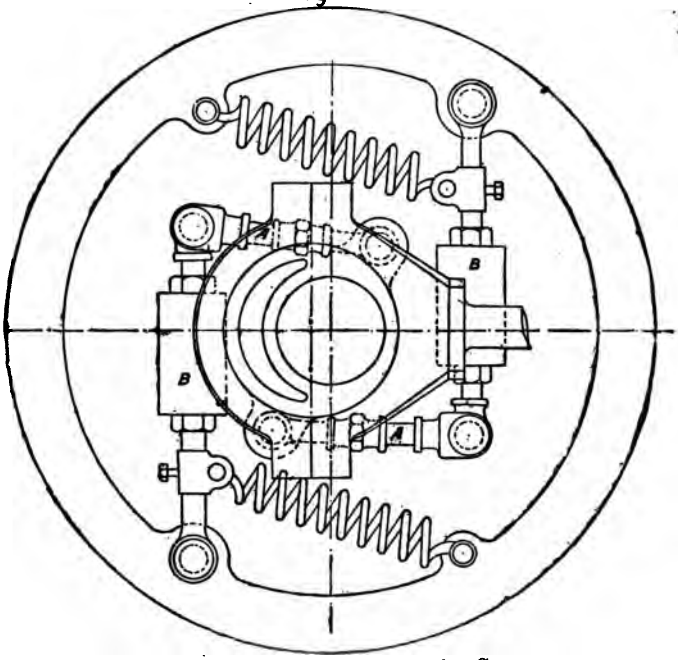
Fig 43.



for this class of gears. O A is the position and throw of the main or distribution eccentric; and O B the expansion eccentric. C D is the lap circle, which, in this instance, is negative. Cut-off is at O E. Suppose B moved round the shaft to position O B′, then O F is the cut-off line.

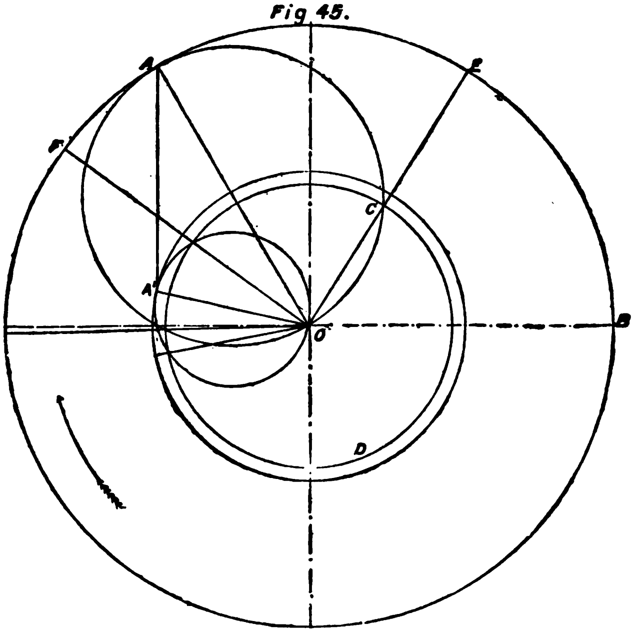
Fig. 44 illustrates a device whereby the angle of O B can be varied according to circumstances. The figure is self-explanatory, and it is only necessary to remark that the design admits of considerable adjustment. In the first place, the point of attachment of the springs to the levers carrying the weights B B can be altered; the position of the weights themselves can be adjusted by

Fig 44



Crank-Shaft Governor Expansion Gear.

Fig 45.



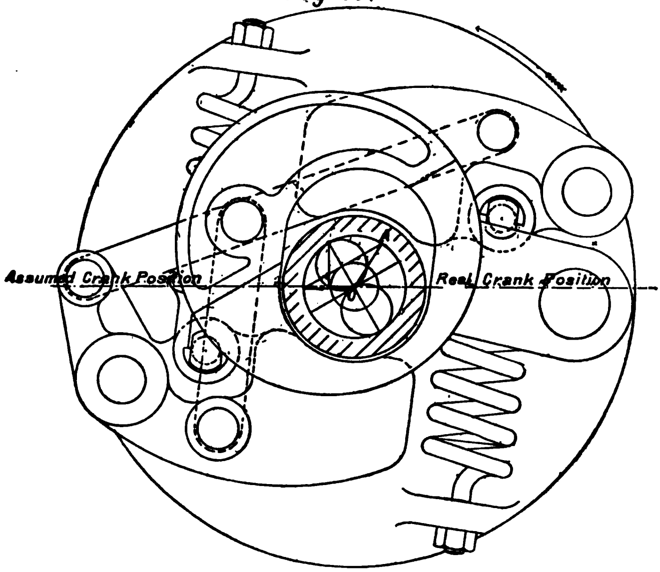
means of the nuts on the levers; and the connecting links A A are also adjustable.

The third method of obtaining a range of expansion is by varying both the throw and position of a simple slide valve eccentric. The principle of this class of gears is set forth in the accompanying diagram.

Let O A (Fig. 45), represent the throw and position of the eccentric relative to the true position of the crank O B. C D is the outside lap circle, and cut-off is at O E. Let the eccentric be altered so that its throw and position are given by the line O A. The lap circle will remain the same, and the cut-off will therefore be at O F.

Westinghouse Governor.—As a typical example of this class of gears, the Westinghouse governor and eccentric is selected.

Fig 46.



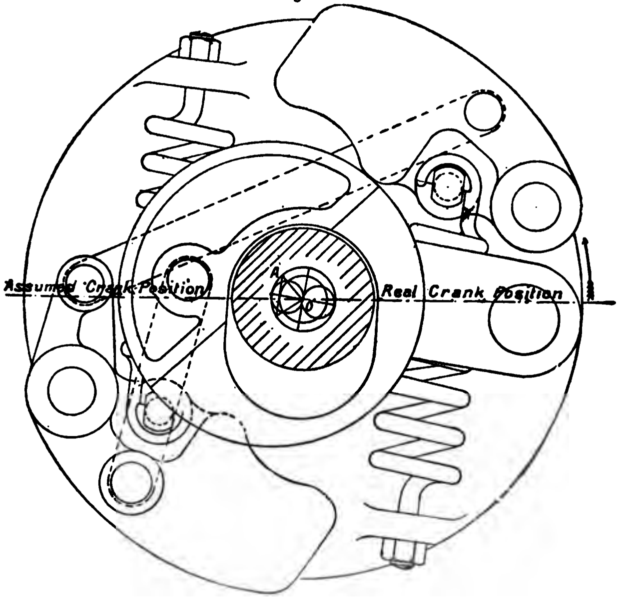
Westinghouse Governor (position for latest cut-off).

Fig. 46 shows the position of the parts for the latest cut-off, which is at O A. In Fig. 47 the gear is in position for earliest cut-off, which is at O A. The real and assumed crank positions are indicated on each figure, as is also the direction of rotation. The valve diagrams show that as the governor moves from latest to earliest cut-off, the lead increases. This governor operates upon a piston valve, which is the most suitable form for crank-shaft governors, as the work required to move the valve passes partly through the springs.

Crank-shaft governors find much favour in America. The

Dick and Church governor, as made by the Phœnix Iron Works Company, is representative of its class.

Fig 47.



Westinghouse Governor (position for earliest cut-off).

The principal forms of slide valve expansion gears have now been dealt with. The varieties and modifications are numerous, but the action is generally little different from the gears that have been described, a knowledge of which will enable the action of other gears to be understood.

CHAPTER III.

LINK MOTIONS.

To enable an engine to run in either a forward or backward direction, it must be fitted with link motion, or some other equivalent device; and it remains to discuss various gears by means of which reversal is effected.

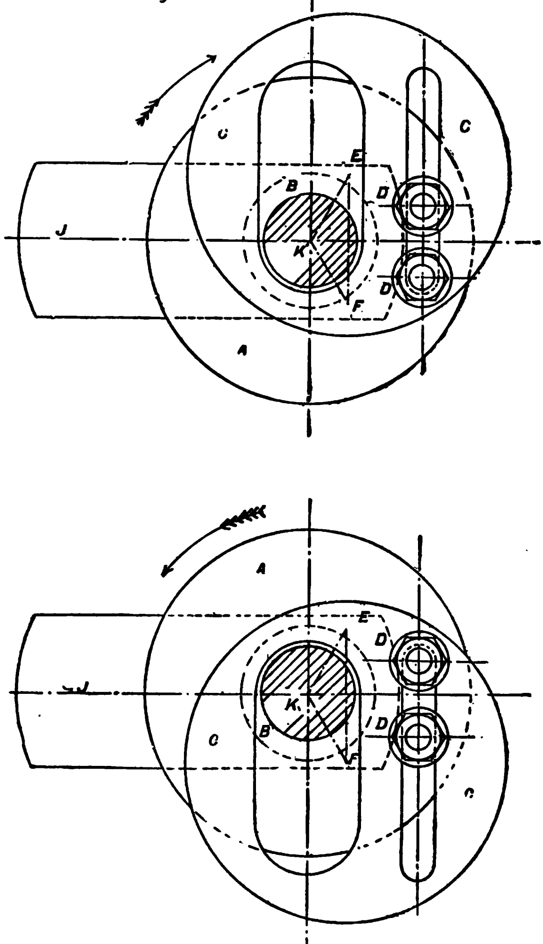
Single Eccentric Reversing Gear.—The simplest form of reversing gear consists of an eccentric, the angle and throw of which can be varied at will by moving it across the shaft. This device, although simple and efficient, would be very inconvenient, especially on an engine of large size, or on one that required frequent reversal; and the arrangement is here described, not because of its practical utility, but because of the principle involved—the principle of link reversing gears.

The single eccentric reversing gear consists of a plate A, secured to the crank shaft B by means of a key. The eccentric C is provided with two slots, whereof the larger is to allow of the eccentric being moved on the shaft, so as to take either the position shown by the upper or lower diagram, Fig. 48, or any intermediate position between the extreme limits. The second slot is made to pass the bolts D D, which are securely fixed in the plate A. This arrangement permits of the eccentric being held in any position between E and F.

In the diagram the position of crank is indicated, and also the centre line of the eccentric. It will be understood that when the eccentric is secured so that it coincides with the line K E—its extreme position in one direction—the engine will run in the direction of the arrow, and when the eccentric has the position K F, the motion is reversed The reason of this is clear. Consider first the upper diagram, where the crank is on dead centre, and the line K E is the position of the eccentric. The cylinder is supposed to lie to the left of the figure. Motion by piston effort

at this position is, of course, impossible, but suppose the crank moved by some means in the direction of the arrow. The eccentric moves through the same angle in the same direction. This movement will manifestly drag the valve spindle, and consequently the

Fig 48.



Single Eccentric Reversing Gear.

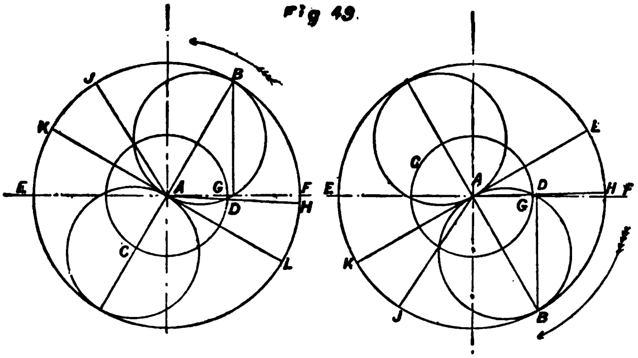
valve, from the back end of the steam chest towards the front end. When the crank is at the back dead centre, K J, the eccentric, if correctly set, will have opened the back steam port for lead, and any movement of the crank in the arrow's

direction will cause the port to be opened wider. The engine would therefore run in the direction indicated, or "over," as it is commonly termed. If the crank were moved in the opposite direction, the eccentric still being held in the position K E, the valve would have the back port open to lead as before, but motion of the eccentric in that direction would close the port, thus preventing the engine running "under."

Now consider the action when the eccentric coincides with the line K F, which position is shown in the lower figure. The crank may be supposed to be on dead centre as before. Any movement in the arrow's direction opens the back port wider; and the engine is therefore free to revolve in a counter-clockwise direction. Moving the crank in a direction opposite to that indicated would close the back port, thus preventing clockwise running.

Valve Diagram for Shifting Eccentric Gear.—The Zeuner valve diagram is well adapted to the solution of problems in

Fig 49.



Zeuner Diagram.

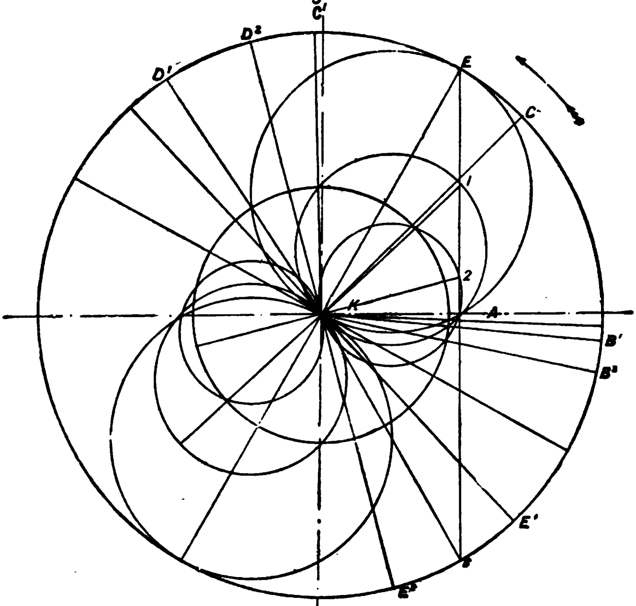
connection with the shifting eccentric gear.* Let A B represent the position of the eccentric when set for running in the arrow's direction. A B in the left-hand diagram of Fig. 49 is, in fact, the line K E of the upper diagram in Fig. 48, and A B in the right-hand diagram (Fig. 49) is the line K F in the lower diagram (Fig. 48). On A B describe the polar valve circle. From B draw B D perpendicular to the horizontal E F, and mark off the lead D G. A G is therefore the amount of lap. Draw the lap circle G C. Admission, cut-off, release, and compression are at A H, A J, A K, and A L respectively.

The inherent peculiarity of the Zeuner diagram is here seen. The *real* position of the crank relative to the eccentric's position A B, is A E; but the steam distribution is traced round in the direction indicated, as though the crank were *leading* the eccentric.

* This matter has already been touched upon when dealing with crank-shaft governor gears.

The state of things for any position of the eccentric between E and F (Fig. 48) can now be shown. K E and K F in Fig. 50 are in length and position equal to K E and K F of Fig. 48. Join E and F. The line E F represents the path of the eccentric sheave when being moved from one extreme position to the other. Wherever the position of the eccentric may be, the line E F must terminate the radius of the eccentric throw, and, therefore, the line drawn for any intermediate position of the eccentric on the line E F, to the centre K, represents by its throw and position the radius and angle of the eccentric sheave; and if on that line

Fig 50.



produced, valve circles be drawn, the action of the steam is rendered clear. In the accompanying figure this has been done. Two intermediate points have been taken and numbered 1 and 2. Admission, cut-off, release, and compression are at B^1, B^2; C^1, C^2; D^1, D^2; and E^1, E^2, respectively.

Tabulating these results it appears that—

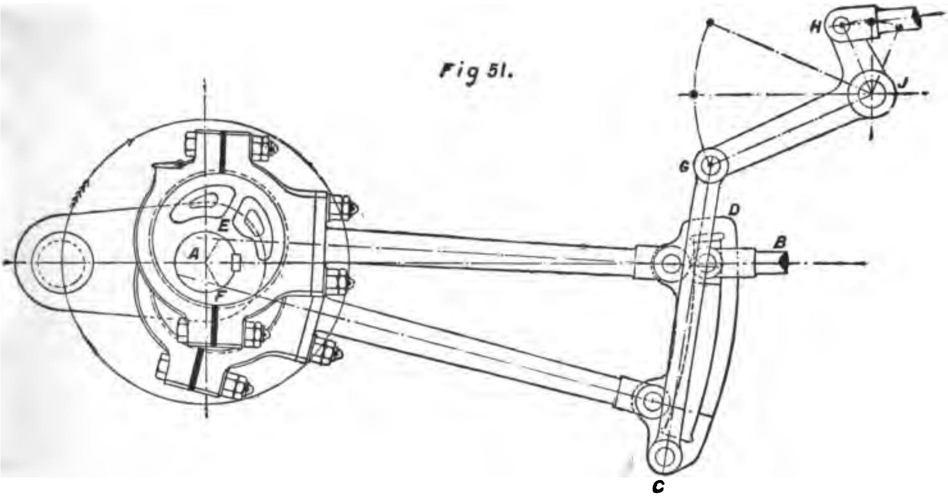
When the eccentric is at K 1	Admission is earlier Cut-off ,, Release ,, Compression ,,	than when the eccentric is at K E.
When the eccentric is at K 2	Admission is earlier Cut-off ,, Release ,, Compression ,,	than when the eccentric is at K 1.

It also appears from the diagram that, although admission is earlier as the eccentric approaches its neutral position K A, the lead is constant for all positions, because the cosine of the eccentric angle is constant. This is true, neglecting the angularity of the eccentric-rod, which is generally so small as to be quite inappreciable.

The line E F will be designated the characteristic line of the gear, and it will be shown that in certain forms of reversing gears it is a straight line, as in the present instance; and in others it has a curvature, sometimes towards the crank shaft, and sometimes towards the cylinder. The significance of the various forms will become apparent as the work proceeds. For the present it will suffice to say that when the characteristic line is straight and vertical, the lead is constant for all positions of the gear.

Stephenson's Link Motion.—Fig. 51 shows the well-known Stephenson, or shifting link, motion. A E is the throw and position

Fig 51.



Stephenson Link Motion.

of the eccentric for running under, and A F the eccentric for running in the reverse direction. The link D C is curved with a radius equal to the length of the eccentric-rods,* and is concave to the crank shaft. The valve spindle B is coupled to the link D C in such a manner that the latter is capable of being raised or lowered

* When the length of the eccentric-rods is spoken of, it should be understood to denote the distance from the centre of the eccentric clips to the centre of the eccentric pin holes on the link, unless there is a direct statement to the contrary. Strictly speaking, the form of link shown in the figure has a radius greater than the length of the eccentric-rods, by the distance between the eccentric pin hole and the pin hole in the link block, when in full gear.

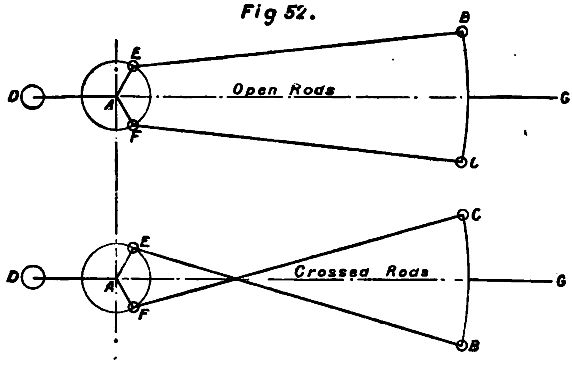
by the lifting link C G, which is in turn connected to the bell-crank lever G H, having J as a fulcrum. This lever may be operated by a reversing handle, a reversing wheel and screw gear, or by a small cylinder fitted for the special purpose. The latter arrangement constitutes what is known as a "steam reversing gear," and is most frequently seen on large winding engines.

In the diagram the eccentric A E has sole control of the valve, the action of which is similar to that of a valve connected to the eccentric A E in an ordinary manner. The crank, therefore, revolves in the direction indicated. The angle and throw of the eccentrics are determined by the construction of a diagram similar to that illustrated in Fig. 1. When the eccentric A F has control of the valve the action is practically the same, but the motion of the crank is reversed.

The action of the motion for full forward and full backward gear being understood, the question arises, what would be the action for any intermediate position of the link block?

Open and Crossed Rods.—Before dealing with this question, the meaning of the terms "open" and "crossed" rods should be

Fig 52.

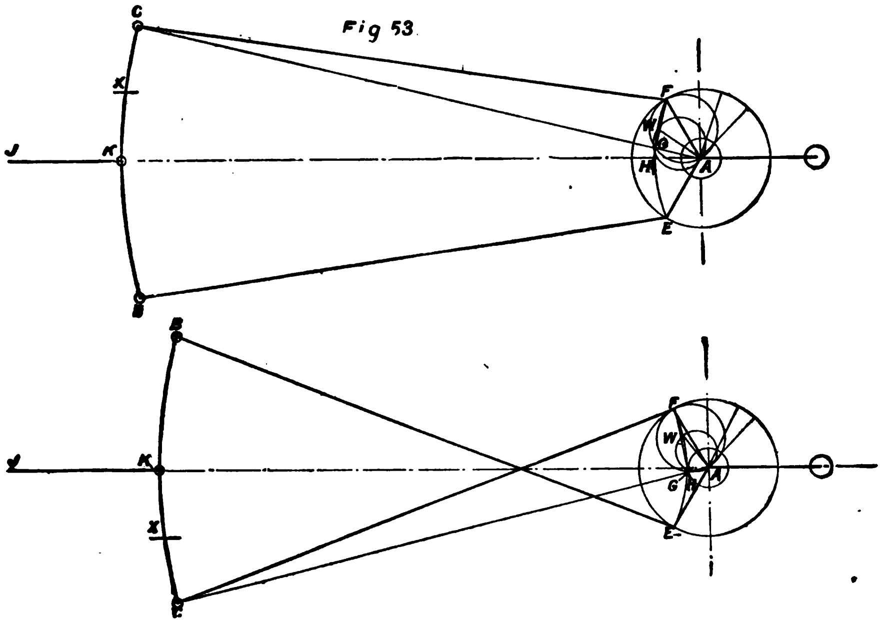


made clear. In Fig. 52 A E and A F are respectively the forward and backward eccentrics of a link motion driving a valve spindle G. In the upper figure the eccentric-rods are coupled to the link on the side on which are situated the respective centres of the eccentric sheaves, when the crank is on the outer dead centre. This arrangement constitutes an "open" rod link motion. In the lower figure the eccentrics are coupled as shown, thus forming what is known as a "crossed" rod link motion.

Valve Diagram for Stephenson's Link Motion.—Let A F and A E, Fig. 53, be the forward and backward eccentrics, respectively, of two link motions, whereof the upper diagram is an open, and the lower a crossed rod arrangement. B C is the link, and F C and E B are the eccentric-rods. On A E describe the valve circle, cutting the line A C, which is drawn from the extremity of the link C, to the centre of the crank shaft in G. From F through the point

G, draw the straight line F H, meeting the horizontal line A J in H. Through the three points E, H, and F draw an arc. This arc will be the characteristic line of the gear, and includes the extremities of all the virtual eccentric arms for any position of the block in the link. Take any position of the gear, and divide the line E F in the same proportion that the block divides the whole length of the line. Referring to the diagrams, let X be the point of gear chosen. It is required to find the movement of the valve corresponding to this particular position. On the characteristic line E F mark the point W so that F W bears the same proportion

Fig 53.



Valve Diagram for Stephenson's Link Motion.

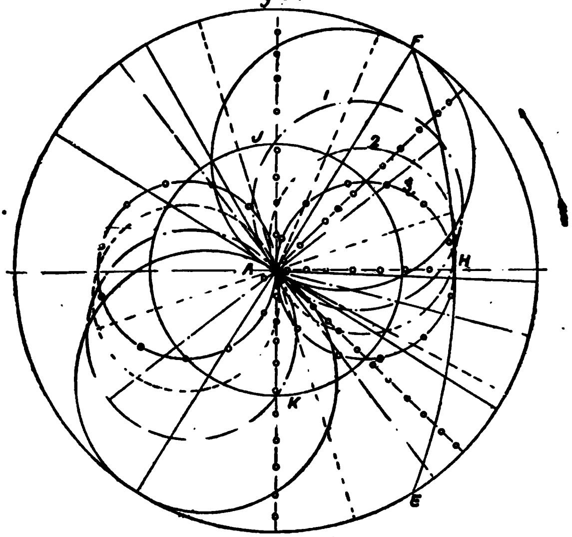
to the line E F that X C bears to the whole length of the link C B. Join the points A and W. Then, A W is the virtual eccentric arm which produces approximately the same valve movement that results from the combined action of the two eccentrics A E and A F, acting through the medium of the link B C. On A W as diameter draw the valve circle. The maximum port opening, the periods of admission, cut-off, release, and compression are now clearly seen; and by choosing other positions of the link and repeating the above construction the action for any other point is seen.

Consider the enlarged diagram for open rods, Fig. 54. The letters A, E, F, and H correspond with the letters of the previous diagrams;

and the characteristic line E F contains the extremities of all the virtual eccentric arms. The lap circle J K remains constant for all positions of the gear; the valve circles, however, are changed for every position of the gear, and cross the lap circle at different points, and, in the manner of the simple Zeuner diagram, indicate the percentages of admission and cut-off. The lead is also seen to vary for all positions of the link, a fact which explains that common statement that in the shifting link gear, with open rods, the lead increases as the gear is brought into mid-position; but with crossed rods, the reverse is the case.

It can be shown that these statements are correct from an inspection of Fig. 52. The motion for both open and crossed rods is shown in mid-gear; the crank in each case is on the outer dead

Fig 54



point, and the port is open for lead. Suppose the link B C moved so that the point B is brought to the centre line D G. The point B is constrained to move in an arc whose centre is E, and therefore the shifting of the link motion must pull the valve to the left, because E is fixed. This movement manifestly lessens the port opening and decreases the lead. Applying the same reasoning to the crossed rod gear, it is seen that the valve spindle moves to the right as the block in the link approaches full gear. Generally, the longer the eccentric rods and the shorter the valve travel, the less will be the variation of lead. The truth of these statements is at once verified by a glance at the figure.

Macfarlane Gray's Method of Describing the Characteristic Line. — Macfarlane Gray has suggested a very

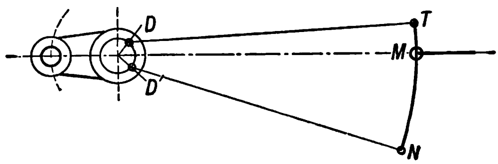
simple method of describing the line E F, which is as near the truth as practice demands. Describe with centre on A J produced (Fig. 53) an arc passing through E and F with radius found as follows:—

Radius = length of eccentric-rod from centre to centre × half the distance between the two eccentric sheaves ÷ the distance between the centres of the eccentric-rod pins on the link. Referred to Fig. 55, Radius $= \frac{DT \times DD'}{2TN}$.

For open rods the arc is concave to the shaft; for crossed rods concave to the link.

It is instructive to apply the two methods given above, to one case of link motion, and observe the closeness of their agreement.

Fig. 55.



Link Motion "notched up."

The above constructions emphasize the advantage of keeping the eccentric throws short, the eccentric-rods long, and the link short, in order that the lead disturbance shall be a minimum. The first and third conditions are to be met by employing double-ported valves; and to keep the eccentric-rods long, the link should be located as near the valve chest as circumstances permit.

At this point it may be well to explain that the diagrams just given are approximate only. When valves are operated directly from eccentrics, the movement can be predetermined with great exactitude, but with link motions it is not so. So many p ints step in to modify the valve movement, that to pre-arrange a link motion that shall satisfy all conditions is impossible. The best that can be done is to secure the best action for full forward and full backward gear, and then, with the link properly suspended, the eccentric-rods long, and the valve travel short in proportion, the action for intermediate gear will be almost exactly that shown by the characteristic line in the diagram, and will be found satisfactory.

The foregoing constructions are, as said, only approximate, but there is a method whereby the valve movement can be determined exactly when the leading dimensions have been settled upon.

Let the accompanying diagram (Fig. 55a) represent, in outline, a link motion in which it is desired to determine the exact movement of the link. Let A E and A F be the backward and forward eccentrics respectively, B C the link, and D G the suspension link, shown in position for mid-gear. Divide the eccentric circle E F into

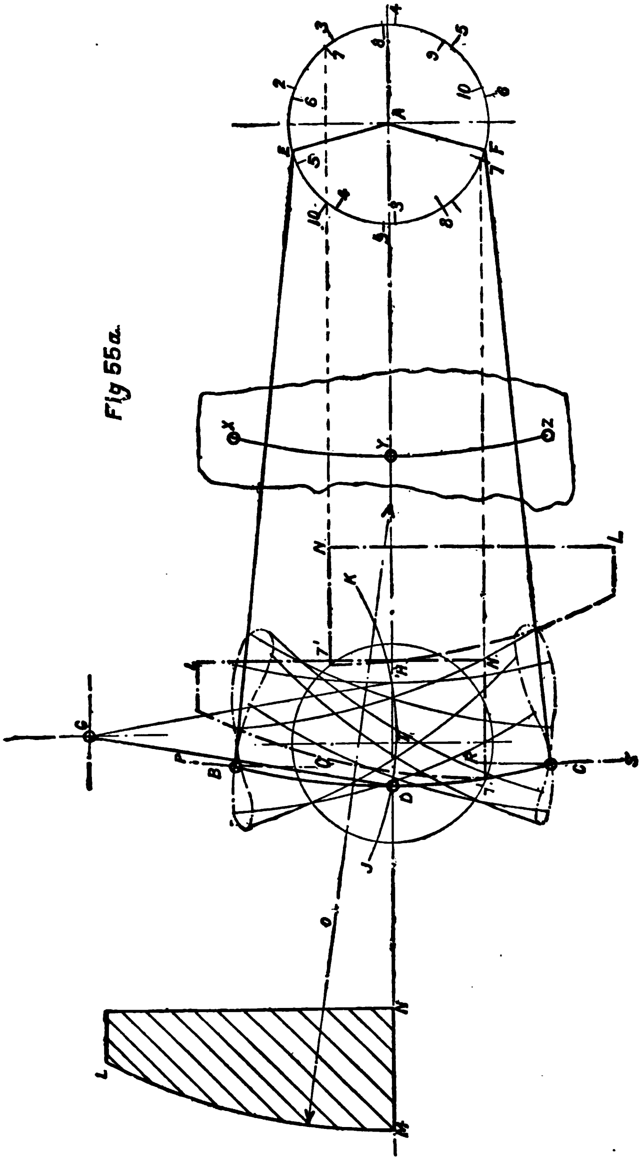


Fig 55a.

any number of equal parts, say ten, starting from the point E for the backward eccentric, and from F for the forward eccentric. (See Fig. 55*a*). From E as centre, and with the length of the backward

eccentric as radius, describe an arc P Q; and from F as centre, and with the length of the forward eccentric as radius, describe an arc R S. Now, it will be apparent that when the eccentrics are at A E and A F, the centre of one pin hole of the link must be somewhere on P Q, assuming that the link is of the double bar form, shortly to be described; and the centre of the other pin hole must be somewhere on R S; whilst the third point is somewhere on the arc J K, the path of the end of the suspension link.

Now prepare a tracing paper templet of the link, drawn to scale, marking thereon the position of the pin holes and the point of suspension. In the figure, such a templet is shown, X being the backward pin hole, Z the forward pin hole, and Y the suspension point. Applying this templet to the arcs P Q, R S, and J K, so that X is on P Q, Y on J K, and Z on R S, a definite position of the link is given, which can be pricked through to the paper below, and numbered to correspond with the eccentric circle. By repeating this process for all the other positions of the eccentrics, the point-path of the pin holes is tracked out, discovering the exact movements of these points. Taking the case of the present gear, the extreme horizontal points of the point D are seen to be D and H; and since the valve spindle is in the direction A D, its travel will be exactly equal to D H; and, further, since the link is in mid-gear, D H is equal to twice the lap *plus* twice the lead, so that either of these being known, the other is found. The peculiar paths of the pin holes are known as "slip curves," and indicate the amount of slotting motion of the die in the link.

The above method of tracing the valve motion is open to one objection, namely, that if the point-paths are to be drawn full size, the drawing must be as large as the whole gear itself, a condition which is usually inconvenient; but, by the method about to be described, the point-paths may be found without drawing the eccentric-rods in position.

In the figure, suppose the eccentric circle E F transposed to the left a distance A T = E B = F C; and let a templet L M N be prepared, having the radius O equal to A T, and the side M N in the direction of the radius. If this templet be applied to the transposed circle, so that the point M coincides with the chosen point in the circle, and the side M N is parallel to A D, the curved side L M contains the centre of the eccentric pin hole. The templet L M N being made of cardboard, or some other suitable material, the curve can be at once drawn on the paper. Similarly, placing the templet in position for the other eccentric, the curved side contains the centre of the other pin hole. Now, applying the link templet as before, so that each centre falls on the curves thus found, the link's position is discovered.* Referring to the figure,

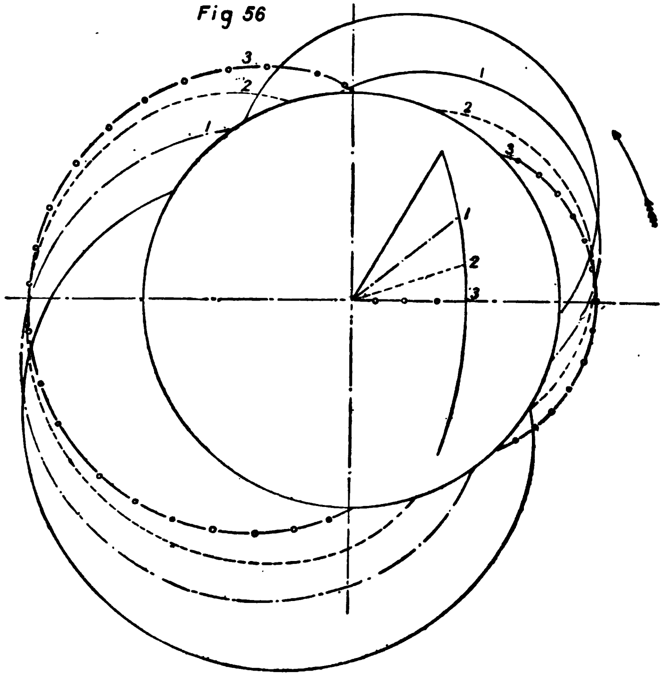
* A valuable paper on "Point-Paths in Mechanisms, found by the use of the templet," appeared in *Engineering*, vol. lvii., p. 439. The use of a templet in this way is applicable to many forms of link-work, and is especially valuable when the mechanism is too large to admit of all parts being drawn in position.

the positions of the templet for backward and forward eccentrics, when the latter are at 7, 7, are shown by dotted lines.

The exact valve movement being thus determined, the results may be referred to an ellipse diagram, or, still better, to a crank pin diagram about to be described. The inequalities for backward and forward running, and for full and middle gear are then clearly shown.

Crank Pin Diagram.—In order to show the steam action for various positions of the link block it is convenient to construct a crank pin diagram giving the opening to steam, the point of

Fig 56



admission, and other events in the stroke referred to a circle representing the path of the crank pin. In the accompanying figure, the curves 1, 2, 3 give the steam distribution corresponding to the valve circles 1, 2, 3 of Fig. 54. From these curves it is clearly seen that as the gear is moved towards the central position, the admission is earlier, the port opening less, and the cut-off earlier. The case in point is one in which inside lap, either positive or negative, is absent. The construction of this diagram, although tedious, is a very simple matter. The Zeuner diagram being drawn, the port opening for any crank position is known, and if various points through the stroke be taken, and the port

openings for each of these positions marked from the crank-pin circle (not necessarily drawn to scale), curves are obtained similar to those shown in the figure. The crank-pin diagram illustrates the action, perhaps, clearer than any other; and it has been the practice of some engineers to have such a diagram plotted down for each gear made, and carefully preserved with the drawings of the engine. The peculiarities of the gear are thus graphically depicted; and should occasion arise to modify the valve motion, there is a true record of the existing conditions, and alterations can be made with confidence, knowing the exact state of matters with respect to the steam distribution.

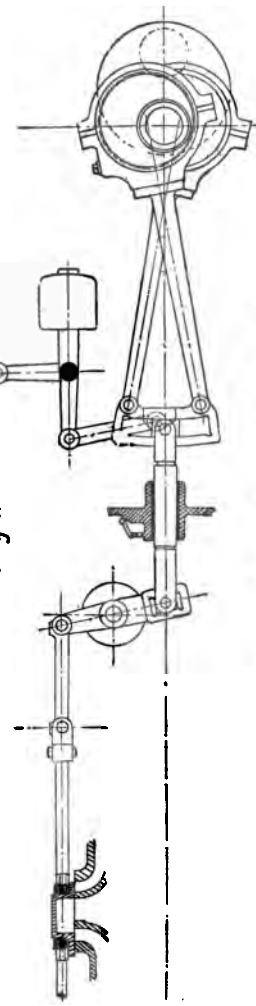


Fig 57

North-Eastern Railway Company's Link Motion.

Best Arrangement of Gear.—In the shifting link motion, the point of suspension C, Fig. 51, may be at one extremity of the link as shown, or at a point midway between the eccentric-rod pin holes. The link for the former mode of suspension is of simpler construction than in the latter case. The suspension point, however, is often decided by the disposition of the adjacent parts of the engine. The central position, generally speaking, gives the easiest motion; but in many cases, notably in locomotives, suspension at this point would mean a very short lifting link, which is an objection. Another point requiring attention is that the arm of the lever G J is parallel to the valve spindle when the link is in mid-gear; and further, when the valve is central on the ports, the perpendicular from the suspension should bisect the versed sine of the arc described by the point G. In the figure these conditions have been fulfilled. Although not of vital importance, the arrangement just described is the best, and wherever possible should be adopted.

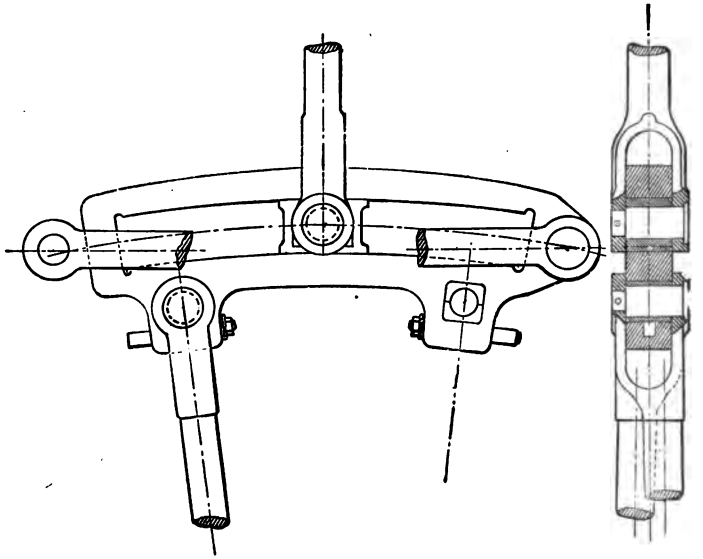
If any link gear be plotted down for various positions throughout the stroke, the centres of the pin holes in the link will be found to describe peculiar curves, which resemble the figure of an attenuated

eight. These are known as "slip curves," and by their breadth denote the slip of the block in the link. The aim of the designer should be to reduce this slip as much as possible, so that the wear of the link is a minimum.

Link Motion made by the North-Eastern Railway Company.—Fig. 57 shows the form of link motion employed by the North-Eastern Railway Company on their express passenger locomotives. The valve chest is on the top of the cylinder, and the valve spindle derives its motion from the rocking lever in the manner shown. The object of presenting this illustration is to show the effect of the rocking lever on the setting of the eccentrics. In ordinary cases the eccentrics would be set in advance of the crank; but here they follow it, and have a position directly opposite the true setting for a link motion of the ordinary type.

The Slot Link.—The ordinary form of slot link is illustrated in Fig. 58.* The point of suspension is on the side remote from the

Fig. 58



The Slot Link.

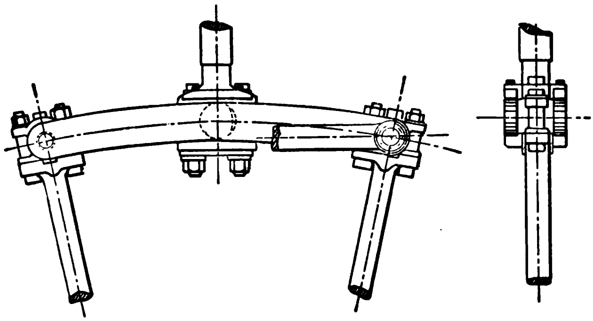
weigh-shaft. This form of link is very simple, and is much used for small engines. Its chief drawback is that no adjustment of the link block is possible; consequently any wear thereof causes considerable jar on the gear.

The Double-Bar Link with Eccentric-Rods Inside.—The next form is the double-bar link with eccentric-rods inside.

* The three drawings of links shown are taken from Seaton's *Manual of Marine Engineering:* Charles Griffin & Co., Limited.

The defect of the common slot link is avoided, but this advantage scarcely compensates for the defect introduced. The link cannot

Fig. 59.

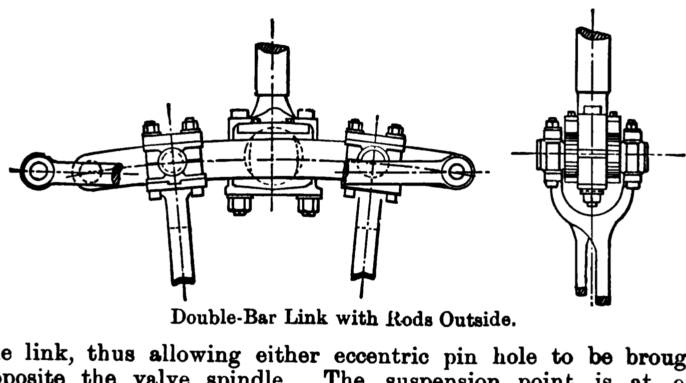


Double-Bar Link with Rods Inside.

be moved so that the end of the valve spindle is opposite the eccentric pin holes, and therefore the throw of the eccentrics must be greater than half the valve travel when set full gear. This is a disadvantage, since it incurs larger eccentrics, and, as a consequence, increased friction. The link itself is also considerably lengthened.

The Double-Bar Link with Eccentric-Rods Outside.—The double-bar link with rods outside is the best form, but the most expensive. The eccentric-rods are here forked, and embrace

Fig. 60.



Double-Bar Link with Rods Outside.

the link, thus allowing either eccentric pin hole to be brought opposite the valve spindle. The suspension point is at one extremity of the link, but, if desirable, could easily be in the centre. The above form is largely adopted in the Navy and Mercantile Marine.

Seaton, in his *Manual of Marine Engineering*, has given rules for proportioning the various details of link motion, which are of great practical value (Chapter xiii.).

Proportions of Links.—For the slot link, let D be the diameter of the valve spindle; then

Diameter of block-pin when overhung	$=D$.
Diameter of block-pin when secured at both ends	$=0{\cdot}75D$.
Diameter of eccentric-rod pins	$=0{\cdot}7D$.
Diameter of suspension-rod pins	$=0{\cdot}55D$.
Diameter of suspension-rod pin when overhung	$=0{\cdot}75D$.
Breadth of link	$=0{\cdot}8$ to $0{\cdot}9D$.
Length of block	$=1{\cdot}6$ to $1{\cdot}8D$.
Thickness of bars of link at middle	$=0{\cdot}7D$.
If a single suspension-rod of round section, its diameter	$=0{\cdot}7D$.
If two suspension-rods of round section, their diameter	$=0{\cdot}55D$.

Distance from centre to centre of eccentric-rod pin holes not less than two and a-half times the travel of the valve when in full gear, and when space permits, two and three-quarters to three times.

The unit D is obtained from the expression

$$D=\sqrt{\frac{L\times B\times p}{F}};$$

Where L = length of valve in inches.
B = breadth of valve in inches.
p = maximum absolute pressure to which valve is exposed, in pounds per square inch.
F = a factor having the following values under various circumstances:—

When the spindle is long and of iron $F=10{,}000$.
When the spindle is long and of steel $F=12{,}000$.
When the spindle is short and of iron $F=12{,}000$.
When the spindle is short and of steel $F=14{,}500$.

For the double-bar link with rods inside let D = diameter of valve spindle, as before.

Depth of bars	$=1{\cdot}25D+\frac{3}{4}$ inch.
Thickness of bars	$=0{\cdot}5D+\frac{1}{4}$ inch.
Length of sliding block	$=2{\cdot}5$ to $3D$.
Diameter of eccentric-rod pins	$=0{\cdot}8D+\frac{1}{4}$ inch.
Diameter of centre of sliding block	$=1{\cdot}3D$.

Distance from centre to centre of eccentric-rod pin holes three to four times full travel of eccentric-rods.

In the double-bar link with rods outside, D = diameter of valve spindle.

Depth of bars	$=1{\cdot}25D+\frac{1}{2}$ inch.
Thickness of bars	$=0{\cdot}5D+\frac{1}{4}$ inch.
Length of sliding block	$=2{\cdot}5$ to $3D$.
Diameter of eccentric-rod pins	$=0{\cdot}75D$.

Distance between eccentric-rod pin holes two and a-half to two and three quarter times the full travel of the valve.

Gooch's Link Motion.—The stationary-link gear, introduced by Sir Daniel Gooch, consists of two eccentrics A E and A F coupled to the link B C by eccentric-rods E B and F C. The link is curved towards the cylinder with a radius equal to the length of

the radius-rod B D; which latter is raised or lowered by means of the lifting link G H. Finally, the link B C is sustained or sus-

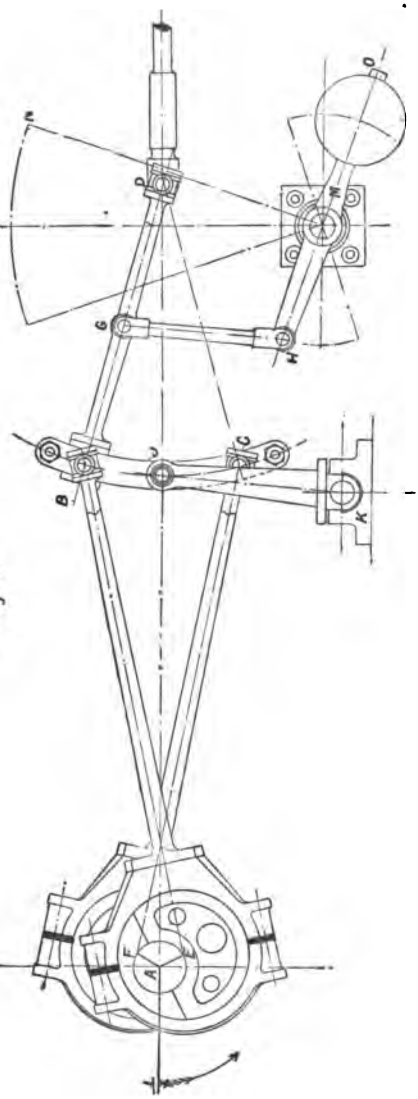


Fig 61

Gooch's Link Motion.

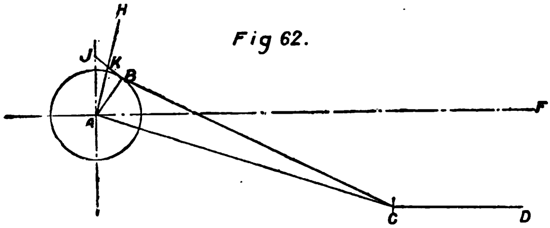
pended by a link J K; K being a fixed centre of motion. Compared with the Stephenson gear, the present arrangement has more joints, and occupies a greater length; but these objections are

compensated in some degree by the improved valve movement, whereby the lead is constant for all positions of the radius-rod. That such is the case will be apparent from an inspection of the figure. The crank A L is supposed to be on the outer dead centre, and when in this position, the valve, if correctly set, gives the desired amount of lead. Now, since the link has a radius equal to the length of the radius-rod, and because the eccentric-rods being of equal length, the link is vertical, it follows that B D can be swept through the whole length of the link without altering the position of D; and since the valve spindle is stationary, the lead is unaltered.

It remains to find the characteristic line of the gear; and, having obtained this, to determine the action for full forward or full backward gear, or for any intermediate position.

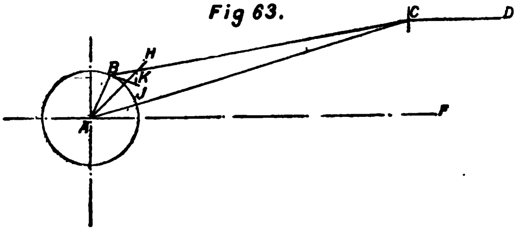
Eccentrics Driving Obliquely.—Before entering on this matter it is necessary to investigate the action of an eccentric

Fig 62.



driving a valve spindle in an oblique direction. Let A B (Fig. 62) be an eccentric, and B C an eccentric-rod driving a valve-rod C D in a horizontal direction. Draw the straight line A C connecting the centre line of the crank shaft and the end of the eccentric-rod. This line makes an angle C A F with the horizontal line A F. Draw the line A H, making angle B A H equal to angle C A F. At right angles to B A draw B J, intersecting the line A H at K. Then A K is the throw and position of an eccentric, which, driving the valve spindle directly, would give approximately the same steam distribution as an eccentric A B driving the valve spindle obliquely. The foregoing construction, although not admitting of

Fig 63.



exact mathematical proof, is so very near absolute truth as to be quite satisfactory for all practical purposes.

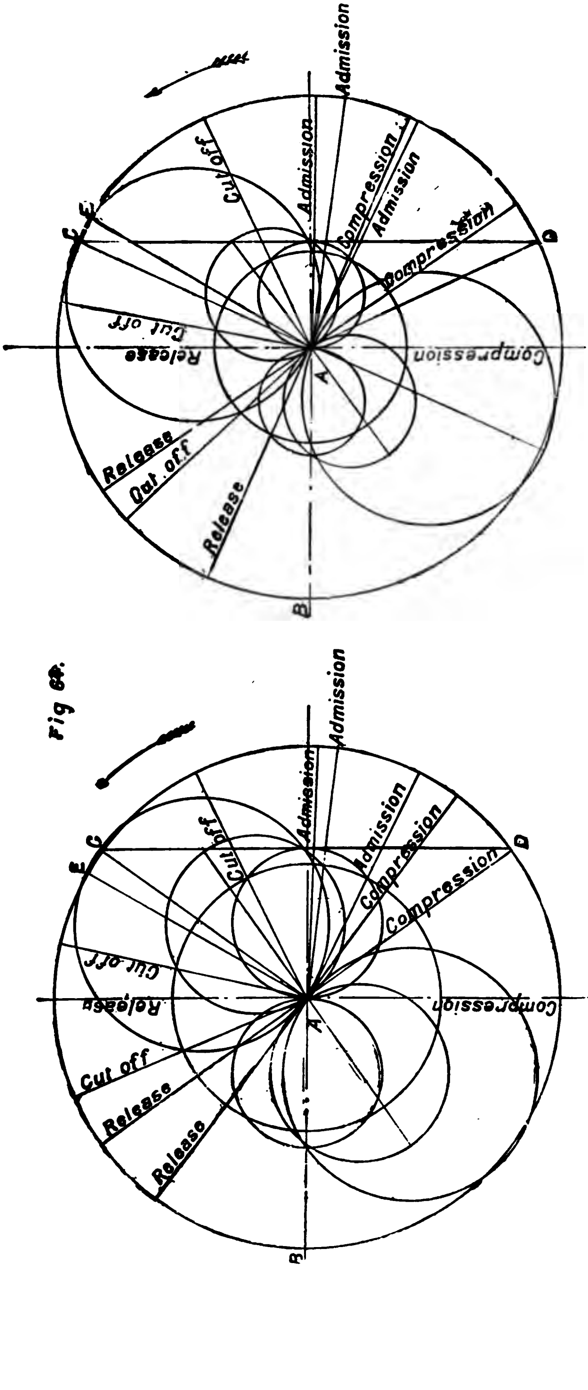


Fig 64.

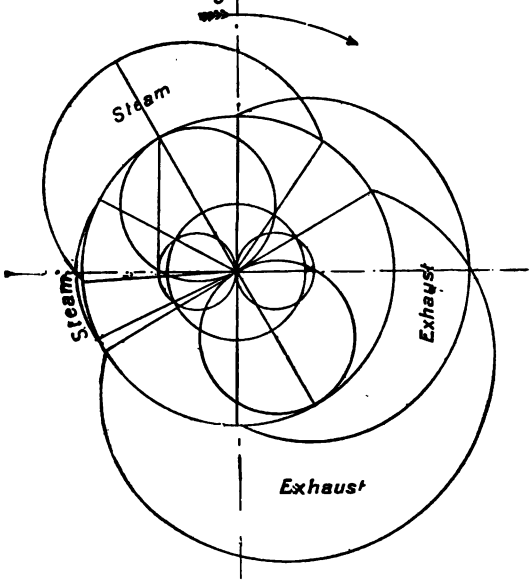
Crossed Rods.

Open Rods

In Fig. 63 consider the eccentric A B driving the valve spindle in the manner shown. Join A C, and make angle B A H equal to angle C A F. Draw B J perpendicular to B A, cutting the line A H at K. Then A K is the throw and position of an imaginary eccentric, which, driving the valve-rod directly, would give approximately the same action as A B driving it in an oblique manner.

Valve Diagram for Gooch's Link Motion.—In Fig. 64 let A E represent the throw and position of one of the eccentrics in a stationary link motion, in relation to the real position of the crank A B. Make A C equal to the throw and position of the virtual

Fig 65.



eccentric arm, obtained in the manner just explained. Draw the vertical C D, which is the characteristic line of the gear, and contains the extremities of the virtual eccentric arms for all positions. By dividing the line C D in the same proportion that the block divides the link, the equivalent eccentric for that position is obtained, and the valve action for that particular position can be shown by Zeuner circles. In the figure several positions have been taken, and the points of admission, cut-off, &c., clearly indicated.

Crank-Pin Diagram.—In figure 65 a crank-pin diagram has been constructed, for full and mid gear, showing, as the characteristic line indicates, a constant lead for both grades of cut-off.

Best Arrangement of Gear.—The stationary link is more sensitive to the point of suspension than the shifting link, and unless there are strong reasons for not doing so, the point of suspension should be placed in the arc of the centre line of the link, and midway between the eccentric pin holes. The correct horizontal position of K is somewhere on a vertical line which bisects the arc described by the central point of the link, J—an arc whose length is equal to twice the lap *plus* twice the lead. Finally, the link J K should be as long as convenient. The above conditions have been carried out in the design shown by Fig. 61. It is difficult to explain why these positions are the best; but plotting down the slip curves for various points of suspension will clearly show that they are.

As to the point G (Fig. 61), it may be anywhere on the radius rod, and is often placed half way between the extremities. The intermediate link-work between G and the reversing handle M N is necessarily one of convenience, the aim being to move the block of the radius-rod its full extent by a suitable movement of the handle on the reversing lever. To arrange for a movement of from 2 feet 3 inches to 2 feet 9 inches is a common allowance. When reversal is effected by a hand wheel and screw thread, as is common in locomotives and marine work, the mechanical advantage between the wheel and the lifting link is not restricted in any way. To balance the weight of the radius-rod and lifting link, the lever H M is made double ended, and carries a balance weight on the arm M O.

The proportions of the Stephenson link given on a previous page apply equally to the Gooch link gear, and can be relied upon giving good results.

Allan's Straight Link Motion.—The Allan straight link motion is obviously a combination of the shifting and stationary link gears. In the present arrangement the link and radius-rod are moved in opposite directions. The eccentric-rods may be open or crossed. With open rods the lead of the valve increases as the link is brought into mid-gear; but with crossed rods the reverse is the case. It will be shown, however, that this variation is somewhat less than in the shifting link with the same length of rods; but as the Allan gear occupies more length than the former, the eccentric-rods are generally shorter; so that finally, the lead variation is little less than with the common form. The two extra joints introduced are an objection; and altogether it is difficult to see that the straight link gear is an improvement on its predecessors. This much, however, may be said; the movement of the link and radius-rod being in opposite directions, the vibration of these parts between extreme positions is less; and, therefore, in confined situations its use may be desirable. Another point in its favour is the straight link, which is somewhat easier to construct than a curved one.

In the skeleton diagram, Fig. 66, A E and A F are the eccentrics relative to the real position of the crank A L. B C is the straight

link connected to the eccentric-rods. The link is supported from a point central between the two pin holes, by the lifting link D G. This rod in turn is centred upon one end of a double-armed lever

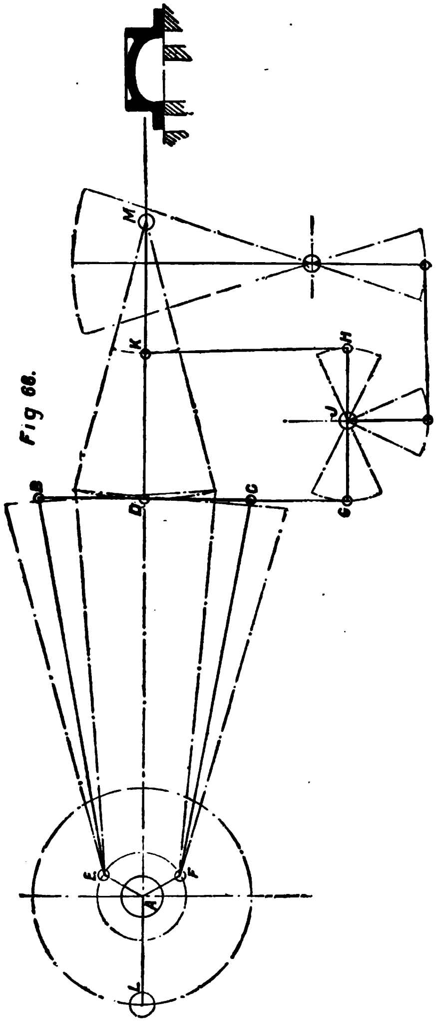


Fig 68.

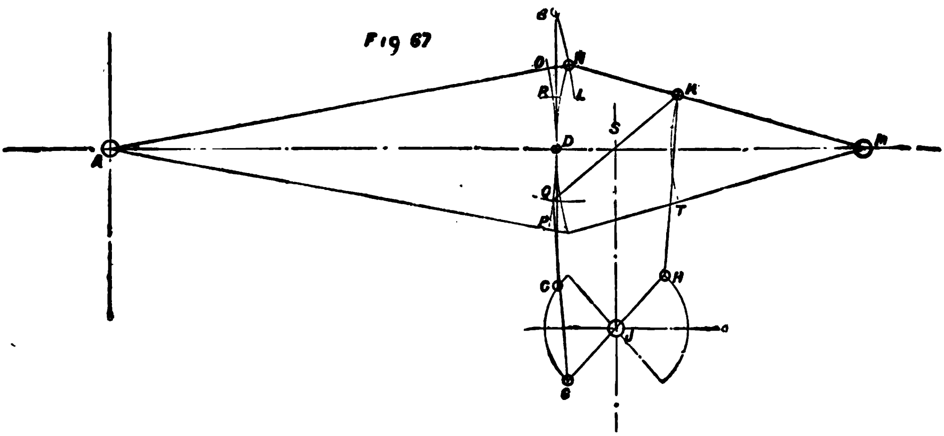
C H having J for a fulcrum. H K is the radius-rod lifting link moving B M in an opposite direction to B C.

In raising the link from mid position to full gear the movement is less than would be the case for the shifting link as said, and the

inequality of lead is approximately that which would result by moving the link of a Stephenson gear, having the same length of eccentric-rods, in amount equal to the distance moved by the Allan link. In the shifting link, it has already been shown that the lead variation increases as the movement of the link, so that the reason of the variation being less with Allan gear will be understood.

Diagram for Allan's Gear.—The location of the centre of the weigh-shaft J is an important matter. In the following diagram, Fig. 67, A is the centre of the crank shaft, and B C is the link. From A as centre, and with A B as radius, describe the arc B L; and from M as centre, and with M D as radius, draw the arc D N intersecting B L in N. Join A N and M N. The point N indicates the position of the ends of the eccentric- and radius-rods when in

Fig 67



full gear. Fix upon the suspension point K, in the radius-rod; and from M as centre and with radius K M describe the arc. From A as centre, and with radius A D, describe arc O P. With centre N, and radius equal to half the distance between the pin holes in link (D B = D C, in the figure), cut the arc O P in Q. Join K and Q. The line K Q crosses the centre line A M at S; and a vertical from S contains the proper centre of the weigh-shaft. Let the vertical position of the shaft be fixed upon, and lettered J. Then the double-armed lever G H should be designed so that the bisection of the versed sines of the arcs described by the extremities of G H are vertical with the bisection of the versed sines of arcs Q R and K T.

The above construction is not imperative, but it is the one that gives the easiest running. Sometimes the point of suspension of the radius-rod is on the opposite side of the link to that shown above, the radius-rod extending past the link B C to accommodate the lifting levers; and sometimes the link is slung from above, the

point of suspension being at the lower end of the link. This is the arrangement adopted by Mr. David Jones on the Highland Railway goods engines.

Expansion Gears and Link Motions Combined.—When expansion gear is employed in conjunction with link motion, the position of the expansion eccentric must be directly opposite the crank, if similar action for forward and backward running is desired.

CHAPTER IV.

OTHER REVERSING GEARS.

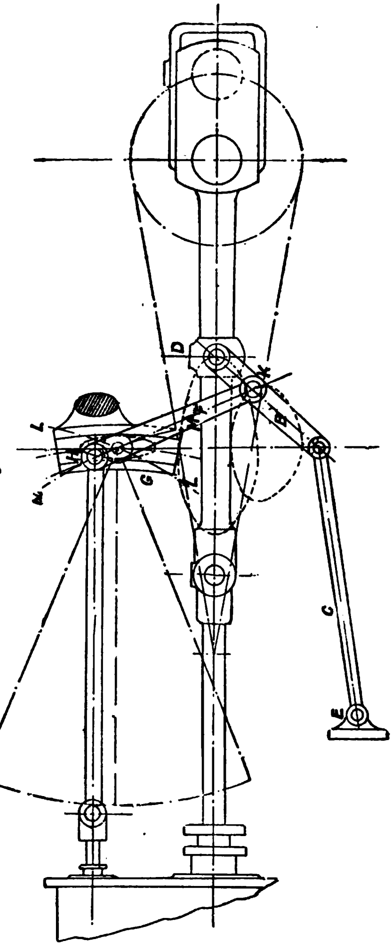
THE defects of ordinary link gears have caused engineers to seek for other means of reversing, which shall be free from the objections to reversing gears of the link type, and which, at the same time, shall be quite as simple and effective. This effort has evolved several well-known forms of link-work for actuating the valve, to which the name Radial Valve Gear is given.

Principle of Radial Gears.—The general principle of radial gears is that of obtaining from some reciprocating or revolving piece of the engine, an arrangement of link-work, a point in which shall describe an oval curve, and by altering the direction of the axes of this curve, to produce reversal, variable expansion, or stoppage.

Joy's Valve Gear.—Of radial gears, the Joy valve motion is perhaps the best known. At a suitable point D in the connecting-rod (Fig. 68), the link B is connected, its lower extremity being guided by an anchor link C, which has E for a fulcrum; E being carried from a suitable part of the engine frame. The link F is guided in a curved slot G, which, together with the movement derived from B, constrains the point H to move in an oval curve. The valve-rod J is connected to the extremity H, and imparts motion to the valve. Plotting down the paths of D, K, and H will discover that these points move in paths similar to the dotted curves shown in the figure. Now the crank being on the outer dead centre, as shown, and the curved slot G being vertical, it will be understood that if the total breadth of the curve described by H be equal to twice the lap *plus* twice the lead, the valve will simply move to and fro on the port face, opening each port alternately by the amount of lead. This will be the action for mid-gear. Now, suppose the angle of the curved slot altered to L L. The path described by H will still be of the same form, but altered in position; and because of this altered position will impart an increased travel to the valve. Again, suppose the slot moved the

same angle on the other side of its vertical position. The valve now has the same travel as before; but, because of the altered angle of the guide, the valve's motion will commence in the opposite direction, thus reversing the engine. It will also be understood that, by giving any intermediate position to the slot between the

Fig 88.



Joy's Valve Gear.

central point and L L, the valve travel is proportionally increased or decreased, as the case may be, thus varying the point of cut-off; and that any intermediate position between the central line and M M will also produce a variation of cut-off; but the motion will be in the reverse direction to that caused by having the slot inclined on the other side of the vertical.

The correct method of laying down the various centres of the Joy gear is set forth in a pamphlet issued by Mr. David Joy, the inventor. The method is as follows:—

Diagram for Setting out Joy's Valve Gear.—On the connecting-rod A B, take a point C, so that its total vertical vibration D D^1 is not less than twice the full valve travel, preferably a little more.* Through D D^1 draw X X perpendicular to the horizontal A B; and at the proper distance from A B lay down the centre line of the valve spindle. Mark the extreme position of the point C for inner and outer dead centres; and choose such a lever C E whose total angle of vibration C E C^1 does not exceed 90°; and carry the end E by an anchor link E F, the mid position of which is parallel to the connecting-rod when horizontal.

Next, on the centre line of the valve spindle produced, and on each side of the vertical X X, mark off the points J J′ each distant from the vertical by an amount equal to lap and lead. From the point J draw a line J H, the centre line of a link that, by virtue of its connection to C E, will move the point K—the point where J H crosses the vertical—equally on each side of the central point K. The point K is the centre of oscillation of the curved guides in which slide the blocks carrying the fulcrum M of the lever J H. The position of H is best found by a tentative process; and to test whether the chosen point be a correct one, the equal vibration required is marked off on each side of K, and lettered L L. The distance L L is equal to the vertical vibration of the point *c* on the connecting-rod, that is, D D′. From D as centre, and with C H as radius, mark the point M in the vertical X X; and with the same radius mark off N from D′. M and N are the positions of H when the engine is at half stroke, or thereabouts, and these points give the total vertical vibration of H. From M as centre, and with H K as radius, describe an arc cutting the vertical X X; and from N as centre, and with the same radius describe an arc also cutting the vertical X X. Then, if the point H be the correct one, the arcs just drawn cut the vertical X X at L and L. Should the points of intersection fall *below* L and L, the point H is too near E; but if they fall *above* L and L, H is too near O. The exact position of H is generally found on a second trial.

The valve-rod J G may be of any convenient length, but the centre line of the slides must be struck with the same radius. From the point K draw a line KO parallel to A B; and with centre on this line, and with J G as radius, describe an arc containing K, and cutting the curves L L, struck from K as centre, in P P′. From P or P′, and on each side thereof, mark off on the arcs L L an amount equal to one and a quarter times the maximum port opening required, and let R R′ be the points. With centres on the arc S S struck from centre K, describe arcs passing through K R and

* The port opening being determined, the desired point of cut-off, and the lead being known, it is possible to estimate the requisite travel by diagram 1 or 5, pp. 3 and 8.

K R′. These arcs represent the centre lines of the curved slots for forward and backward gear, and when the latter are in either of these positions the point of cut-off is about 75 per cent. Should a

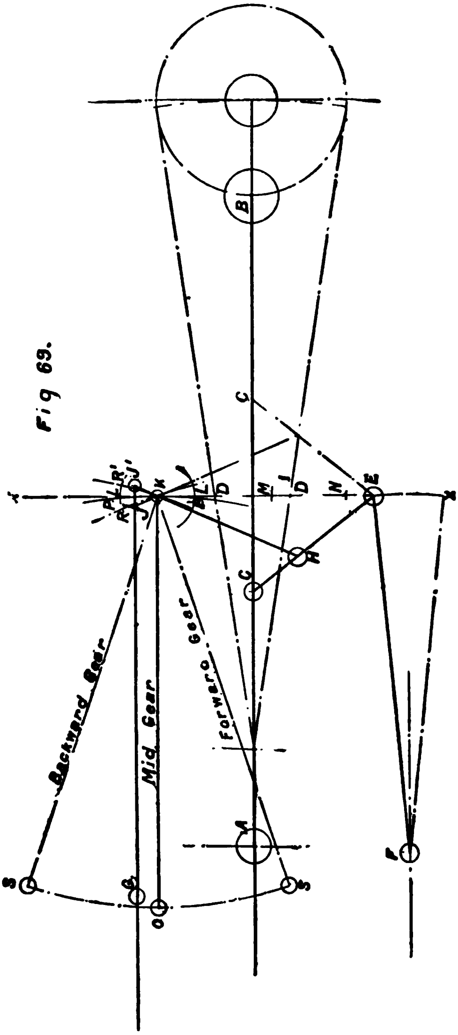


Fig 69.

later cut-off be desired, the slots must be carried still further from their vertical position.

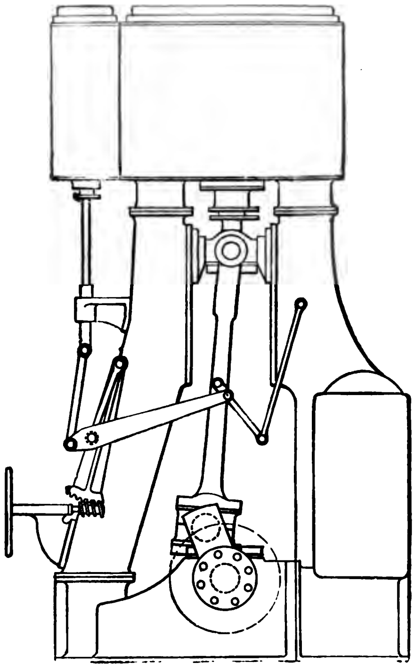
Glancing at the diagram, it will be seen that the fulcrum K of the lever J H coincides with the centre of oscillation of the curved

slots or guides when the crank is on either dead centre. It is evident that when these points coincide the angle of the guide can be altered to any extent without disturbing any other part of the mechanism, a state of things which shows that the lead is constant for all positions of the reversing handle.

The inventor's pamphlet goes on to say that when the above directions are followed, the leads and cut-offs for each end of the cylinder, for backward and forward gear, are practically equal.

The arrangement of the gear as just described is the most effective, but considerable latitude is permissible. For instance,

Fig. 70.



Joy's Valve Gear applied to Marine Engines.

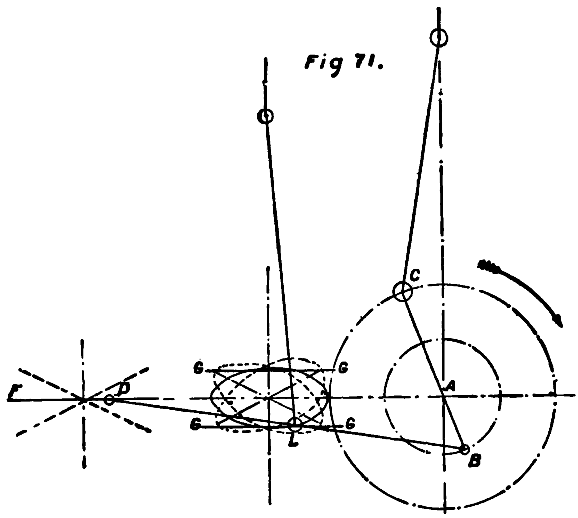
the point C can be placed above or below the centre line of the connecting-rod, and the point K can be raised or lowered, so that the line K O is no longer parallel to A B; but it is not advisable that the line should have a greater inclination to A B than 4°, or thereabouts. Again, the anchor link may be dispensed with, the point J being guided in a slide affixed to some convenient part of the engine.

For vertical engines the same rules apply by placing the diagram vertically, and altering relatively the terms "vertical" and "horizontal."

The valve motion arising from the link-work of this gear is superior to the ordinary eccentric motion. If the movement of the parts be traced round for a complete revolution of the crank, it will be found that when opening and closing the steam ports the valve travels rapidly; but during expansion and exhaust the movement is slow. Another advantage, in the case of locomotives, is the space saved on the crank shaft, due to the absence of eccentrics, whereby longer journals and crank pins are obtainable.

Joy's Gear on Marine Engine.—As an example of the gear applied to a marine engine, the annexed figure (70) from Seaton's *Manual* is presented. The reversing wheel actuating the worm gearing with the toothed sector is clearly shown. The curved guide is here dispensed with, the end of the reversing sector carrying a lever which has the same function.

Hackworth's Valve Gear.—Hackworth's Valve Gear, shown diagrammatically in Fig. 71, consists of an eccentric A B placed



Hackworth's Valve Gear.

directly opposite the crank A C, actuating the eccentric-rod B D, the extremity of which is constrained to move in a straight guide F, the angle of which can be varied similarly to the guide of the Joy gear. A point L in the eccentric-rod is employed to give the valve movement. It will be seen that when the position of the guide is horizontal, the point L will describe an oval curve, whose major axes is parallel to F. When the direction of the guide is altered the axes of the curve vary correspondingly, and the motion imparted to the valve is increased. The gear is so set out that when the crank is on dead centre the point D coincides with the

centre of oscillation of F. It is clear, therefore, that F can be pulled over in any direction without disturbing the lead.

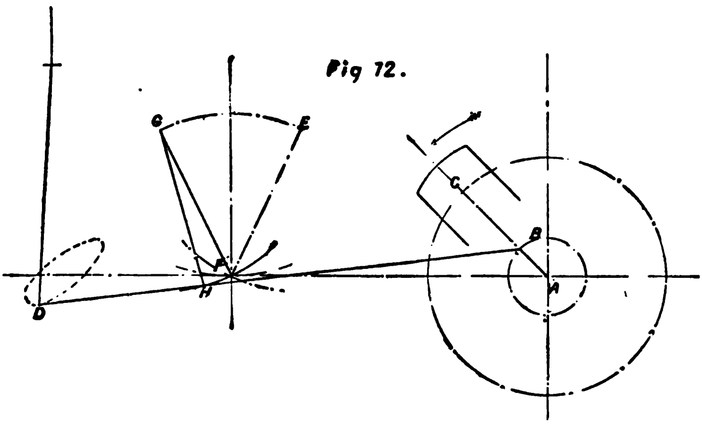
The paths of L corresponding to the three positions of the guide F are shown. Let the horizontal G G be drawn on each side of the horizontal A B, and removed from it a distance equal to the outside lap. The port opening for each position will then be indicated; and from the figure it appears that when the slot is horizontal, the maximum port opening is only equal to the lead.

By tracing the gear round through a complete revolution it will be found that the valve acts most efficiently, having a rapid motion when opening and shutting, but during the closed periods the motion is somewhat slow.

Bremme's Valve Gear.—In the Bremme Gear an intermediate point of the eccentric-rod moves in a curved path instead of a straight line, and the point of attachment of the valve lever is at the extremity of the rod. Fig. 72 will render the action clear.

Diagram of Bremme's Gear.—The eccentric A B, whose position coincides with the crank A C, actuates the eccentric-rod

Fig 72.



Bremme's Valve Gear.

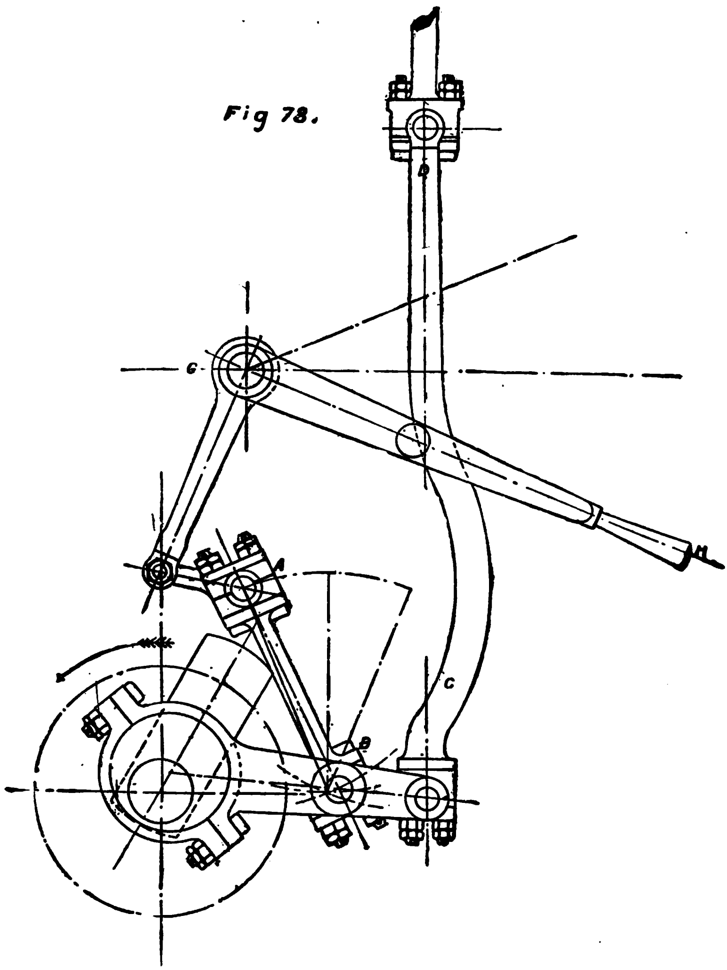
B D, which, at a point H, is constrained to move in an arc having a radius equal to the suspension lever G H, and a centre at G. When the lever G H has the position shown, the crank revolves in the direction indicated. Now let the reversing lever G F, having a fulcrum at F, and the same length as G H, be pulled over to E. The point G of the lever G H now coincides with E, and the arc in which H is constrained to move has its centre at E. This changes the direction of the axis of the oval curve described by D, and consequently produces reversal of the engine.

As in the gears already described, the mechanism is so set out that the guided point of the eccentric-rod coincides with the centre

of oscillation of the reversing lever each time the crank is on the dead centre, thus producing equal leads. The angle of the reversing lever G F from the central line, and known as the "deviation" angle, should, the inventor states, never exceed 25° on either side.

The fact of the point H moving in an arc produces inequality in the valve's motion on each side of its central position, which causes cut-off and compression to fall earlier in the down than in the up stroke; but so far from this being an objection, it is an advantage, for the increased compression serves to neutralise the downward momentum of the piston-rod and connecting-rod.

Fig 78.



Bremme's Valve Gear applied to Vertical Marine Engines.

Practical Example of Bremme's Gear.—Fig. 73 shows a form of the Bremme gear in its practical application to vertical marine engines. The reversing handle G H moves the reversing arm by means of a wiper rod and short drag link. The suspension-rod A B is slung from the extremity of the reversing arm, and carries the eccentric-rod at its lower end. In the figure the reversing arm has been purposely omitted in order to avoid confusion with the suspension-rod A B. The centre line of the former, however, is drawn, and also its extreme position on the other side of the vertical, the total angle of deviation in this instance being 40°. The valve-rod C D and the reversing handle are in front of the engine, the latter being locked in any position by an arrangement carried on one of the engine stanchions.

Comparison of Joy's and Bremme's Gear.—Comparing the merits of the Joy and Bremme gears, it would appear that for locomotives the Joy gear is to be preferred; but Bremme's gear is the best adapted for marine work. It would be somewhat difficult to arrange the Bremme gear with its eccentric-rod and reversing arm underneath the locomotive boiler, so as to be compact and clear the various parts. In marine work, however, the case is different. Here the space for the eccentric-rod and levers is usually abundant, and the reversing lever can be conveniently situated. The movement of the parts of the Bremme gear is considerably less than that of the parts of its rival, and this is a consideration in the engine room of a steamship where space is limited, and where it is necessary for the attendants to be in close proximity to the working parts.

The objection to radial gears, as a class, is the number of parts in continual wear. These parts, unless very substantial and adjustable, soon work loose, and cause rattle, besides disturbing the valve action. But given good design and first-class workmanship, there is no reason why radial gears should not wear as long as link motions, and the former have undoubtedly considerable advantages in the direction of improved steam distribution.

PART II.

CORLISS VALVES.

PART II.

CORLISS VALVES.

CHAPTER I.

GEARS WITHOUT TRIP MOTION.

CONTENTS.—Advantages of Corliss Gears—Classification of Corliss Gears—Valve Gear without Trip Motion—Effect of Wrist Plate—Reduction of Dwelling Angle—Location of Wrist-Plate Pin—Rules for Diameter of Valves—Angle of Vibration—Position of Levers—Valve Diagram for Corliss Gear—Proportions of Steam Valve and of Exhaust Valve—Double-Ported Valves—Equilibrium Corliss Valve—Crank-Shaft Governors applied to Corliss Valves—Diagrams of Valve Motion.

ABOUT the year 1850, G. H. Corliss constructed the form of valve which bears his name. Since that time the Corliss valve and its operating mechanism have been modified and improved, until they have attained their present state of perfection, and seem to have settled down to certain designs, which it appears difficult to better.

Advantages of Corliss Gears.—The Corliss valve may be briefly described as being of cylindrical form, working on a cylindrical face, and vibrating on its longitudinal axis. It is the outcome of an effort to produce a valve gear which would present the following advantages:—

1. Minimum of power to actuate the gear.
2. Smallness of clearance spaces.
3. Adaptability to close governing.
4. Good steam opening; instantaneous cut-off; and free exhaust.

Classification of Corliss Gears.—It will be convenient to classify Corliss gears under three heads:—

1. Gears without trip motion.
2. Single eccentric gears with trip motion.
3. Double eccentric gears with trip motion.

The first class is seldom used. The valves have a positive connection with the eccentric, the travel is constant, and the point of cut-off invariable.

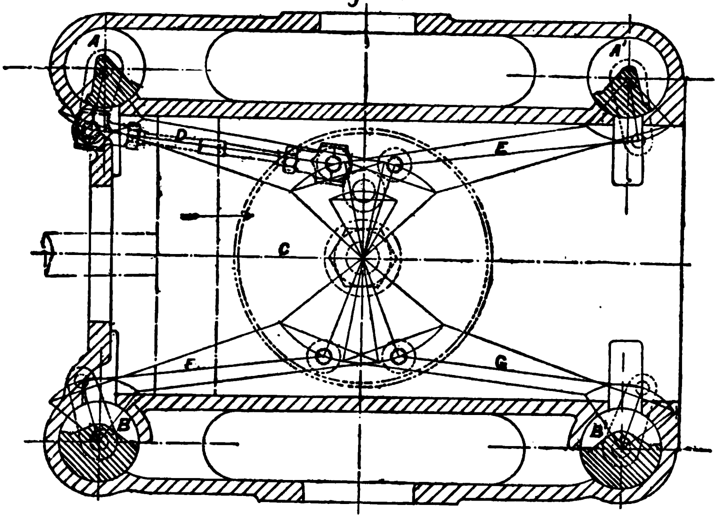
In the second class, the steam valves have no positive connection with the eccentric, but are opened by catches against the resistance of a dashpot, and released at some point of the stroke determined by the governor, or otherwise. The exhaust valves are positively connected to the eccentric and receive a constant movement.

The double eccentric gear is the form usually made in this country. In this class the steam valves are operated in the same manner as in the single eccentric gears, but the exhaust valves are driven from a separate eccentric.

As the action of the first class embodies that of gears with trip motion, it will be conveniently discussed at the outset.

Valve Gear without Trip Motion.—Fig. 74 shows an arrangement of Corliss motion without trip gear. A A′ are the

Fig 74.



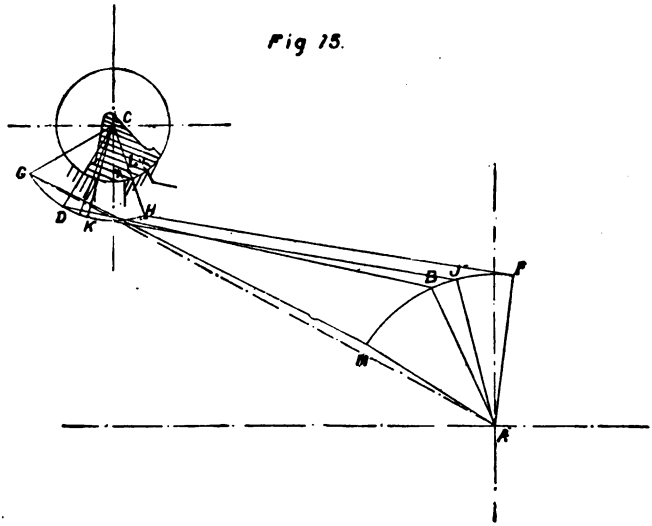
Corliss Motion without Trip Gear.

steam, and B B′ the exhaust valves. These valves are actuated by the eccentric through the medium of the wrist plate C and the wrist plate rods D E F G. When the piston is at the beginning of the stroke and about to travel in the direction of the arrow, the valve will be open a certain amount. This will be lead. At the same time B′ will be open for exhaust, and the valves A′ and B will be closed. The figure shows the wrist plate in its central position, at which time all the ports are closed as indicated. It will be remembered that when a slide valve is in its central position both steam and exhaust ports are closed, assuming that there is no inside clearance; and the probable position of the piston would then be about 10 per cent. from the end of the stroke. The same

condition of things occurs in Corliss gear when the piston is at the same point of its stroke, a position indicated by the dotted lines in the figure. Now it will be easily seen that the front steam valve A has the same function as the front steam edge of an ordinary valve, the back steam valve corresponds to the back steam edge of the slide valve; whilst the front and back exhaust valves produce the same distribution as the front and back exhaust edges of the slide valve. Bearing this in mind the action of the four valves is at once understood.

Effect of Wrist Plate.—The presence of the wrist plate modifies the motion of the valves due to the eccentric to a considerable extent; and the location of the valve-rod pin on the wrist-plate pin is one of the important points in Corliss work. In Fig. 75 is shown

Fig 75.



Steam Valve Lever and Wrist-Plate Rod in three positions.

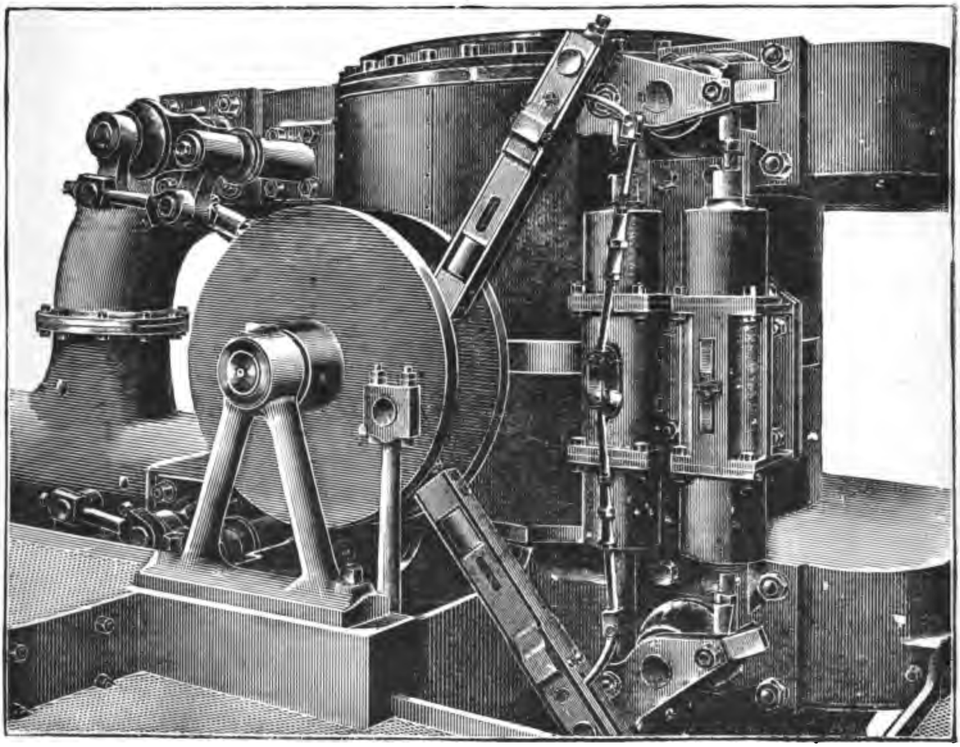
the centre lines of the steam valve lever and wrist-plate rod in three positions. The wrist plate, by virtue of its connection with the eccentric-rod, oscillates through an angle of 33° on each side of its central position, denoted by the line A B; but, by the particular location of the wrist-plate pin B, the valve lever C D swings through an angle of 28° on one side of the position C D, and 51° on the other.

The advantage of this translation of the eccentric's motion will be explained. When the wrist-plate lever is at G, the valve covers the port by an amount equal to the lap *plus* the angle D C G taken at the valve circumference. Now, so long as the valve opens to steam at the proper instant, it matters not what its motion between G and D may be, because all that motion, so far as this particular

valve is concerned, is useless. This movement is known as the "dwelling" motion; for during the time it is being worked through, the valve is merely idling over the port. From what has been said, it is apparent that the less the dwelling angle the better; and one advantage of the wrist plate is this, that it permits of the dwelling angle being kept small. If there were no wrist plate—the eccentric-rod coupling directly to the valve lever O D—the dwelling angle would be half the total angle moved through by the valve.

Reduction of Dwelling Angle.—In very large engines, the advantage of a small dwelling angle becomes more pronounced, and,

Fig 76.



Low-Pressure Gear of Engine designed by Hick, Hargreaves & Co. (by permission of *Engineering*, Ltd.).

cognisant of this, Messrs. Hick, Hargreaves & Co., Bolton in one of their large vertical engines, have gone so far as to introduce an additional lever and rod to actuate the exhaust valves; and by

this means to reduce the dwelling angle to a considerable extent. The accompanying illustration (Fig. 76) is a perspective view of the low-pressure gear of the engine referred to; from which a good idea of the motion may be gathered. On the exhaust side (the left-hand side of the figure) is seen the idling lever and its connection with the wrist plate. Without at present attempting to describe the steam portion of the figure, it may be stated that the trip motion is the well-known Inglis and Spencer gear; and that the large size of the valves necessitated the use of double dashpots; an arrangement whose advantages will be appreciated when trip gears have been discussed.

Another advantage which the wrist plate possesses is the quick steam opening that can be obtained by a suitable location of B (Fig. 75). In Fig. 75, the lap of the valve is indicated, and also the position of the valve lever when the port is open for lead. The corresponding position of the wrist-plate pin is J. The angle J A F is 22°, and the angle K C H, 35°. From K to H is the opening angle of the valve, and it will be understood that the quicker this angle is worked through, the better the steam line on the indicator diagram. Angles J A F and K C H are both worked through in the same time, so that the port opening is quicker in the proportion of $\frac{35}{22}$: 1, than would be the case if there were no modification of the eccentric's motion. The rapid movement through the angle K C H throws little extra work on the wrist-plate pins and rods; for when the valve is moving through the dwelling angle, the whole of its surface is under the influence of unbalanced steam pressure; but when the port is open and an equal pressure is established in the cylinder and valve chamber, the only surface is the width of the facing on the back edge of the port. This distance is indicated by L in Fig. 75.

Location of Wrist-Plate Pin.—Whilst endeavouring to keep the dwelling angle small, it must not be forgotten that by placing the wrist-plate pins too near the horizontal centre line of the wrist plate, the motion will give a double stress on the wrist-plate and valve-lever pins. This should be avoided. In Fig. 75, the centre line of the wrist-plate rod in its extreme position is above the line passing through the extreme position of the valve-lever pin and the centre line of the wrist plate; thus showing that no double reversal of stress occurs. In all wrist-plate gears, the line A M should be the limit of the wrist-plate pin's movement on that side of its central position. The location of B is also governed by the end of the valve-rod, it being evident that the pins must be sufficiently wide of each other to clear the ends of the rods. Sometimes, however, one pin on the vertical centre line of the wrist plate has been used for both rods.

Rules for Diameter of Valves.—The diameter of valves for any given size of cylinder is the next point to consider. This size is usually determined from some empirical formula. It is customary

to let one size of valve serve for several sizes of cylinders, thus effecting a saving in templets and patterns. The rule

$$d = \frac{D}{8} + 2$$

expresses very closely the relation between the size of cylinder and valve. Here d and D are the diameters of the valve and cylinder respectively.

The exhaust valves are sometimes made larger than the steam valves. Then

$$\frac{D}{6} + 2$$

would give a result conformable with modern practice.* The length of the ports on the valve face is in nearly all cases made equal to the diameter of the cylinder, and, having the length decided upon, decides the width.

The length of the valve lever is usually proportioned from the diameter of the valve, although in some cases it is governed to a certain extent by the trip motion. A rule agreeing well with practice and allowing an increasing leverage for high pressures is as follows:—

Length of lever = 1·7 × radius of valve + ⅛ inch for every 10 lbs. above 100 lbs. initial pressure.

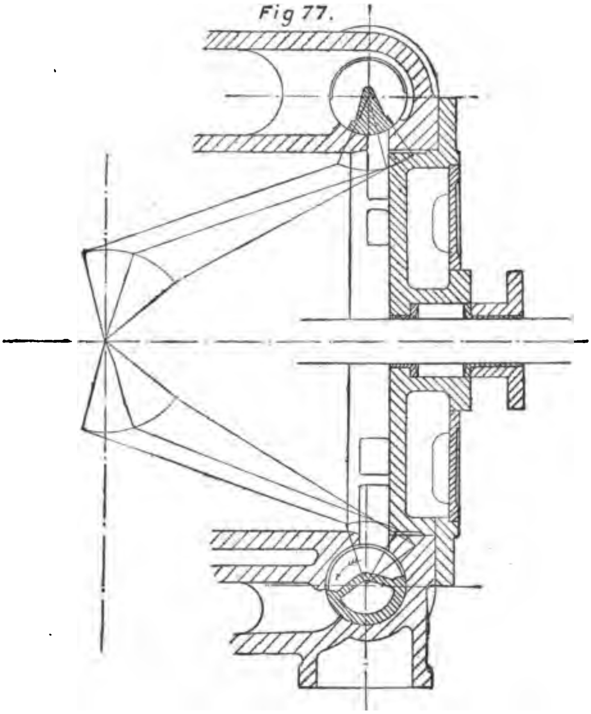
Angle of Vibration.—The angle of movement of the valve levers is usually made from 60° to 70°. The dwelling angle *plus* the lap angle has to be worked through before any port opening is obtained, so that the remaining movement is left for port opening. If the wrist-plate diagram has been laid down, the movement of the valve on the face is at once seen; and when the lap angle has been indicated, the amount of travel left for port opening is shown. It is advisable that this should slightly exceed the width of the port in order that a full port opening may be obtained early in the stroke. Should the angle shown not be great enough to give the over travel, a little alteration in the position of the wrist plate pins would perhaps effect all that is required; care being taken that the angles which the valve lever and wrist-plate rod make with each other are easy. In any case it is not advisable to make the angle between these points greater than 140° on the one extreme position, and less than ~~50°~~ on the other. When these limits are reached in each direction, the angle moved through is about 90°, which is rather excessive. Nevertheless, continental engineers have frequently exceeded these limits, without experiencing any trouble in working. If the necessary movement cannot be obtained except by exceeding the above-named limits, the valve will have to be increased in diameter, or else a double-ported valve must be used. It should be stated that none of the above rules will suit all cases. The wrist-plate diagram is more one of convenience than of exact construction. If all the

* These rules are from the *Mechanical World Pocket Diary*.

angles are easy, and the valves give the necessary port opening, the designer may be satisfied as to its satisfactory working.

Position of Levers.—In arranging the exhaust-valve levers it is necessary to consider the position of the port. Two arrangements are shown in the following figures. In Fig. 77 the exhaust port is towards the centre of the cylinder. In opening, the valve moves in the direction of the arrow, and to give this action at the proper period the exhaust lever must have the position shown by the sketch. In the second case the exhaust port is remote from the

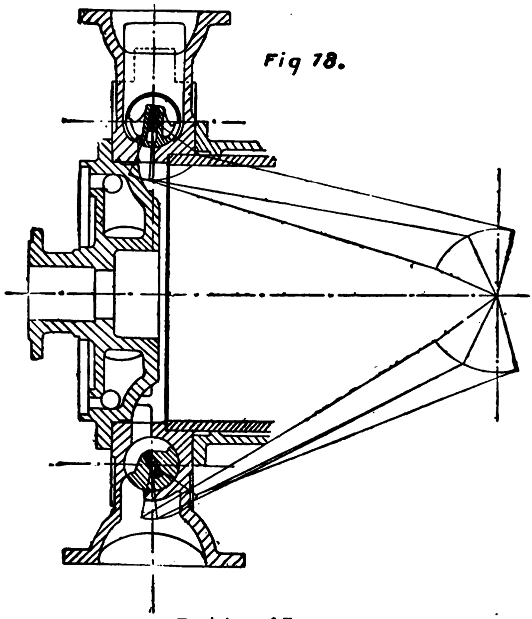
Fig 77.



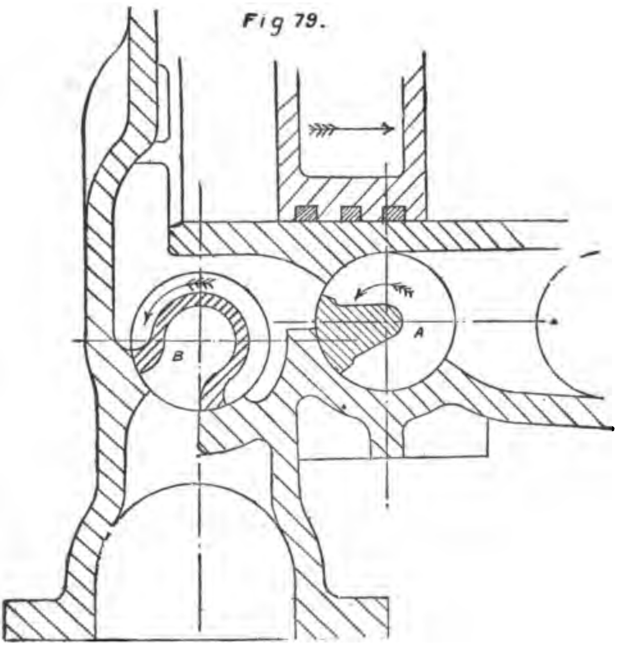
Position of Levers.

centre of the cylinder. This necessitates the valve lever being placed as indicated. Whatever the position of the port, the exhaust valve must be moving in the closing direction when the steam valve for the same end of the cylinder is opening.

Some Corliss engines have both steam and exhaust valves situated below the cylinder. This arrangement, although perhaps giving rather less clearance than the ordinary disposition of valves, is open to the objection that one port serves for both live and exhaust steam; a state of things not agreeable to the modern notion of economical working. The accompanying sketch, Fig. 79, shows



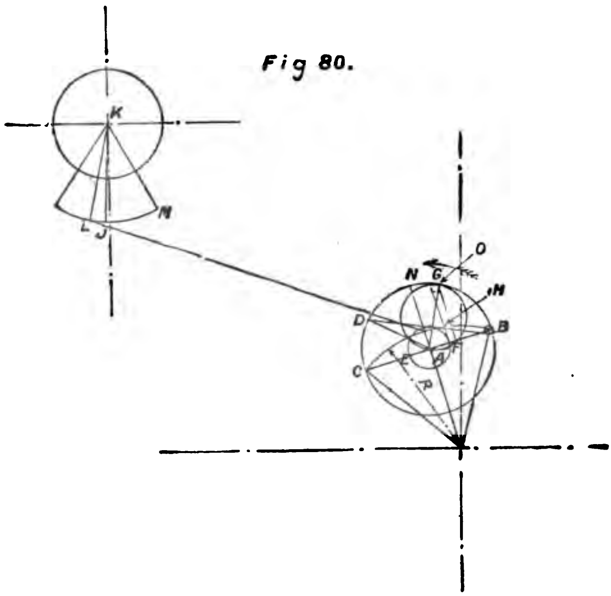
Pusition of Levers.



Arrangement of Steam and Exhaust Valves.

the valves disposed in this manner, wherein A is the steam, and B the exhaust valve. The latter is of the double-ported type in order to reduce the angular movement. The exhaust valve is arranged to drain the cylinder in a very efficient manner and thus obviating the use of drain gear. The relative positions of steam valve, exhaust valve, and piston are indicated, and the arrows show the direction of movement at this particular position.

Valve Diagram for Corliss Gear.—The amount of steam lap required to cut-off at any given point of the stroke is obtained in the manner shown by Fig. 80. Here R is the radius of the wrist-plate pin, and its central and extreme positions are marked A, B,



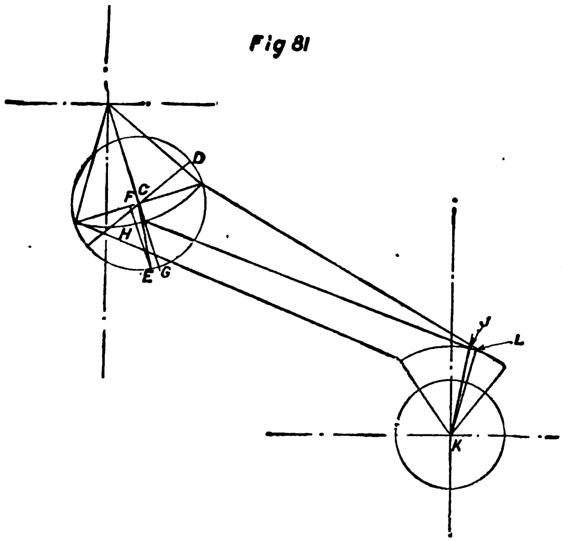
Valve Diagram for Corliss Gear.

and C. Draw the line B C connecting the extreme positions, and on this line as a diameter describe the circle. Imagine this circle to represent the path of the crank pin. Let it be required to cut-off when the crank is in the position A D, which in this instance is about 87 per cent. From B as centre, and with a radius equal to the lead, describe the circle. The line D B drawn as a tangent to the lead circle determines the lap circle E F. At F erect a perpendicular to meet the circle in G. The line F G crosses the radius line of the wrist-plate pins at H. H is the position of the wrist plate pin when the valve opens. From H, and with the length of the wrist-plate rod as radius, draw an arc cutting the valve lever radius at J. Then, in order to cutoff at the required point, the lap

at the circumference of the valve must be equal to the distance between the mid position of the valve lever K L, and the line K J, taken at the circumference of the valve. When the wrist plate is in its central position, both steam valves will cover the port by the amount given by the foregoing construction, and this amount is the lap of the valve.

This construction is faulty, insomuch that it assumes the radius of the lead circle on the line B C. The lead at the circumference of the valve may be known, but this does not give the amount of motion on the line B C to produce it. It is known that the valve lever is somewhere between K L and K M when giving lead; and it is easy to assume a position of the valve lever at lead, and find the requisite motion of the wrist plate to give that amount, and

Fig 81



Exhaust Valve Diagram.

thus get the lead circle on B C very near the truth. Then the diagram may be constructed with this approximate amount; and should it be found inaccurate, a second trial will give a result sufficiently approximate for all practical purposes.

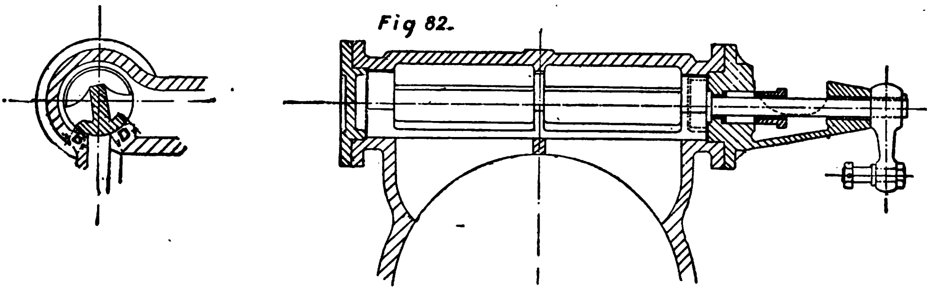
Fig. 81 shows the construction of the exhaust valve diagram. The motion of the wrist-plate pin is indicated, and also that of the exhaust valve lever. Connect the extreme positions of the wrist-plate pin by a straight line; and on this line as diameter describe the circle. This will represent, as in the steam diagram, the path of the crank pin. Now the angular advance of the eccentric is given in the steam diagram, Fig. 80, and is indicated by the letters N A O. This angle being known, the position of the eccentric for

any position of the crank is also known. Let O D represent the position of the crank at release. The corresponding position of the eccentric is C E; the angle E C D being equal to the angle O A C, Fig. 80. Draw E F parallel to G C, cutting the wrist-plate pin arc in H. From H as centre, and with the length of the exhaust valve rod as radius, describe an arc cutting the valve lever arc in J. Draw K J, which will be the position of the valve lever when release occurs. The central position of the lever is K L, and therefore the distance between the lines K J and K L on the valve circumference indicates the amount of exhaust lap that is necessary to produce release at the pre-determined point. In the present instance the lap is of a positive character, and the exhaust valve closes the port by the amount obtained, when the wrist plate is in its central position. If the line K J fell on the opposite side of K L, there would be clearance or negative exhaust lap, whilst the coincidence of these lines would denote that the port and valve edges were edge and edge, when the wrist plate was central.

If the length of the exhaust valve wrist-plate rods is adjustable, as it ought to be, the equalisation of release or compression for front and back strokes is a simple matter. This is most conveniently done when the engine is erected, and the eccentric coupled up. The exact lengths of the rods can then be obtained, and indicated by centre marks thereon

Proportions of Steam Valve.—Fig. 82 shows the construction of a Corliss steam valve, and also indicates the proper dimensions

Fig 82.



Proportions of Corliss Steam Valve.

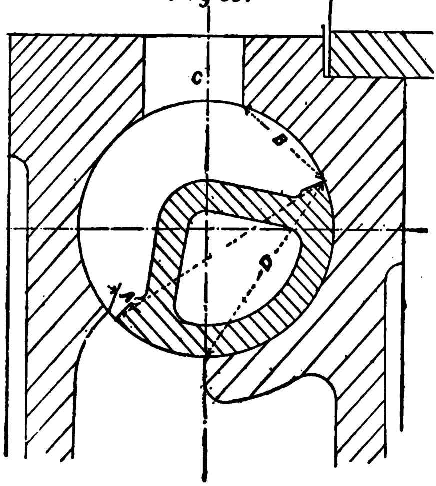
of the face. The distance B is equal to the lap as found by the construction of Fig. 80. The dimension D must be sufficient to lap the back edge of the port when the valve is in its extreme position. D must therefore be equal to the dwelling movement taken at the valve circumference, *plus*, say, $\frac{1}{2}$ inch to insure tightness. The connection of the spindle with the valve is made by a palm forged on the spindle, and fitting into a recess at the end of the valve. This construction allows the valve to bed to its face and follow up any wear, and at the same time permits the valve spindle to work in its bearings without constraint. The spindle is shown provided with a stuffing box and packing gland.

7

This is the most usual construction, but in many cases the packing is dispensed with altogether; the valve spindles being made to work in long bushes accurately bored to suit them. In practice, this method of construction is very satisfactory, and valve spindles thus fitted have been known to run many years without permitting a breath of steam to escape. It is certainly the more mechanical arrangement, and when once in order requires no attention. The friction of a stuffing box is always an unknown quantity, and at times may become excessive and cause undue stress on the pins.

Proportions of Exhaust Valve.—A section of an exhaust valve is shown by Fig. 83. The negative lap is indicated by A. B must not be less than the active angle of movement taken at the valve circumference, otherwise the valve will close the port C, and

Fig 83.



Section of Exhaust Valve.

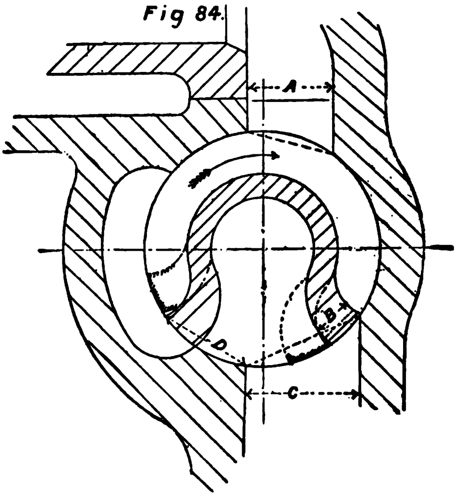
the exhaust will be throttled. D should be a little more than the movement of the valve circumference during dwelling period, in order that the valve may not re-open the port by its back edge.

Double-Ported Valves.—If desirable, the exhaust valves may assume the double-ported form. This type permits of a considerable reduction of travel compared with the single-ported valve. With most English makers it is customary to make both steam and exhaust valves the same diameter, and when the angle of movement for the steam valves is great, it may happen that the requisite angle for the exhaust valves will be still greater and exceed the limits for safe working. In cases of this sort a double-ported valve is convenient.

The annexed figure illustrates the double-ported Corliss valve; the correct dimensions for the face are as follows :—A is the width of

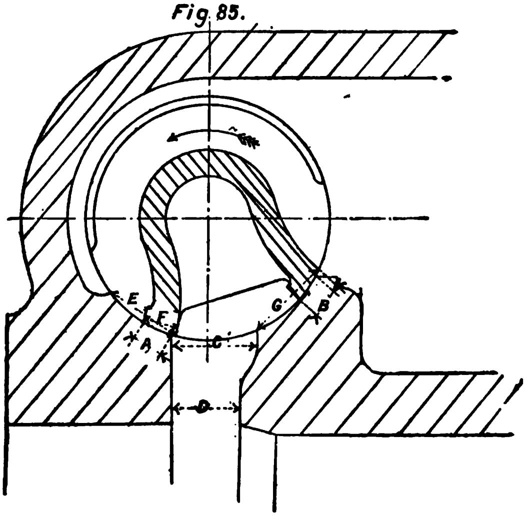
the port in the cylinder. The width of the valve face B is empirical. Then the width of the exhaust port C = A + B. D not less than

Fig 84.



Double-Ported Corliss Valve.

Fig. 85.



Proportions of a Double-Ported Corliss Valve.

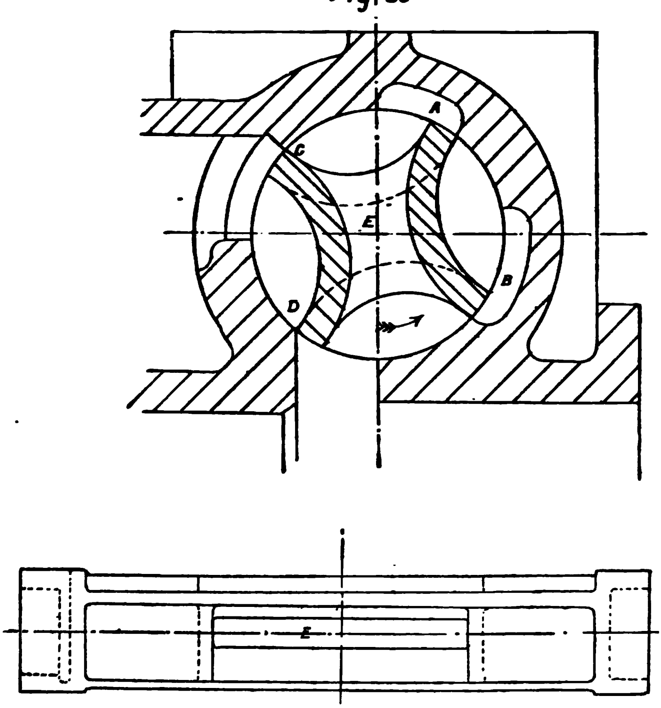
the movement of the valve circumference during dwelling angle, *plus* an amount for tightness. The arrow denotes the direction of

the opening movement, whilst the dotted lines show the position of the valve when full open. The valve should be designed to take up as much of the chamber as possible without wire-drawing the steam, so as to reduce clearance.

The double-ported form may, of course, be applied to the steam valves, and this is often done in large sizes. The form and proportions of such a valve are here shown. A and B are equal to the lap as found by construction. The width of the port at D is what would be necessary for a single-ported valve; but at C the width is D + F, so as to maintain a full opening. E and G should be equal to the movement of the valve circumference during dwelling angle, *plus* a suitable amount for tightness. The width F, as before, is determined by practical considerations.

Equilibrium Corliss Valve.—In order to still further reduce the travel the form of valve shown by Fig. 86 may be employed.

Fig 86



Equilibrium Corliss Valve.

In opening, the movement is in the direction of the arrow, steam being admitted at the openings A and B as well as at C and D.

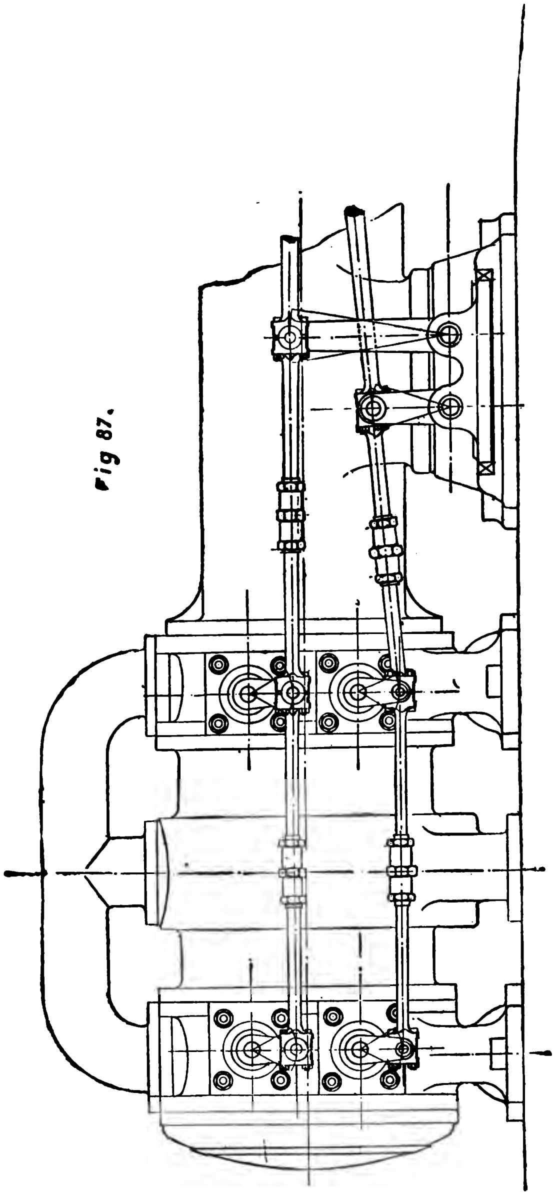
The opening E extends for half the length of the port in the cylinder, as is shown by the elevation view. It is strange that this valve is so little known, or, if well known, so little employed, for it has another advantage besides that of reduced travel. An inspection of the figure will show that at all points the valve is in perfect equilibrium, and that the only resistance to motion is caused by the friction due to the weight of the valve and the friction between the valve spindle and its stuffing-box. It may be objected that because of the peculiar form any wear that takes place cannot be taken up; but where there is no friction there can be little wear, and in practice the valve has proved a success. Its use entails rather more clearance than the ordinary type, but if the valves were placed at the ends of the cylinders, instead of at the top and bottom, in the usual manner, the clearance would still be less than that of most Corliss engines of the present day. It should be added that an engine fitted with this form of valve on the high-pressure cylinder was exhibited at the Paris Exhibition by MM. H. De Ville-Chatel & Co.

Crank-Shaft Governors applied to Corliss Valves.—The combination of a crank-shaft governor and Corliss valves forms a very good arrangement, but one which has not found much favour in this country. The action of crank-shaft governors having been discussed in the first part, further description is unnecessary. It may be necessary to state, however, that the governor must alter both the angle of advance and the throw of the eccentric, in order that the lead shall be constant. A governor of the form shown in Fig. 44, Part 1, will not do; but the Westinghouse governor, Figs. 46 and 47, Part 1, is particularly suitable. It is advisable to employ a separate eccentric for the exhaust valves, so that compression may not be excessive when the engine is running light. The separate eccentric also relieves the governor to a considerable extent.

The annexed illustration shows an engine cylinder with Corliss valves; the cut-off being controlled by a crank-shaft governor. In this case, the valves are located in the covers, and are of the balanced type just described. Separate eccentrics for the steam and exhaust valves are employed; and wrist plates, as will be noticed, are dispensed with; each valve receiving its motion direct from a rocker arm to which the eccentric-rod is connected. This valve gear is specially suited for high-class quick-running engines. It has the simplicity of slide valve gearing without the concomitant defects of the latter; and, for quick running, has advantages over ordinary Corliss gears which will be fully understood when trip gears have been dealt with. The steam branch is below the cylinder; steam passing through a belt cast with the cylinder to the "E" pipe, and thence to the steam valves. The exhaust branches are below, and serve as feet by means of which the cylinder is secured to the foundations.

Diagrams of Valve Motion.—The accompanying diagrams

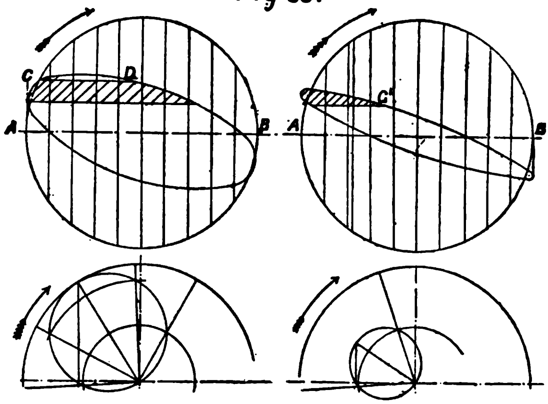
Fig 87.



Engine Cylinder with Corliss Valves.

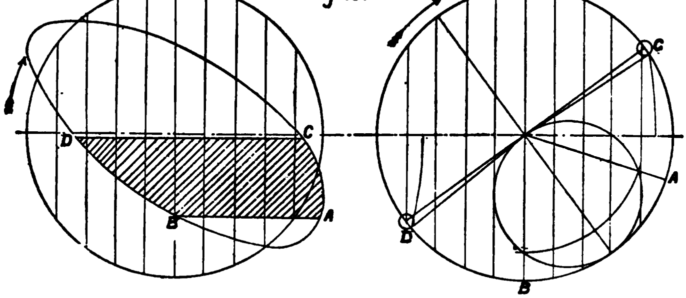
illustrate the action of the steam valves. The horizontal axis A B represents the stroke of the engine; whilst the extreme width represents the travel of the valve. The stroke is divided into ten equal parts, and the points at which these divisions intersect the

Fig 88.



diagrams show the distance of the valve from its central position. The shaded portion shows the port opening, and the fact of the diagram passing beyond the line C D denotes that the valve has over travel. In the left-hand diagram the valve has its maximum travel, and the point of cut-off, C, is latest. At the earliest cut-off,

Fig 89.



the diagram assumes the form shown by the right-hand figure. The lap, of course, remains the same, but the travel is reduced, and cut-off is consequently at C′. Below the ellipse diagrams are shown the steam distributions for the early and late cut-off by Zeuner's method.

The diagrams for the exhaust valves are shown in Fig. 89, from which it will be seen that the valves have over-run; consequently, the port is full open at A, and remains so, until B is reached. Release and compression are at C and D respectively.

This description concludes the remarks on Corliss gears without trip motions.

CHAPTER II.

SINGLE ECCENTRIC GEARS WITH TRIP MOTIONS.

Principle of Trip Gears.—The class of Corliss gears fitted with trip motions has now to be considered. In mechanisms of this description, the steam valves are not in positive connection with the eccentric, but are moved by it against the resistance of a spring, weight, or vacuum. The eccentric-rod, or some other rod connected thereto, is provided with an arrangement whereby it engages with the valve lever, and communicates its motion to the steam valves in the opening direction only. This engagement is broken at a certain position in each revolution, determined by the governor, or by hand, and the valve, freed from all connection with the eccentric, is at once closed by the springs or whatever may be employed for the purpose.

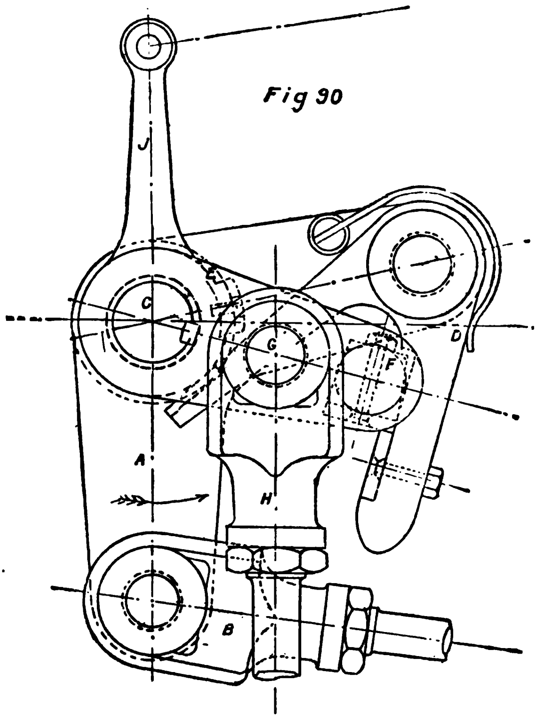
Advantages of Trip Gears.—The arrangement just described has several important advantages. The first and most valuable is the way in which the governor controls the engine; cutting off steam at the point requisite to maintain a uniform speed, and doing so in a manner that relieves it of almost all influence due to the valve; the function of the governor being, not to close the valve, but to determine when it shall be closed.

The second advantage that trip gear possesses is the practically instantaneous cut-off that results from its operation. The steam valves, as previously remarked, are opened against a resistance, which, at the proper instant, closes them with a rapidity and precision altogether unattainable by any motion derived from a reciprocating or rotating part of the engine. This cut-off, indeed, is so rapid, that an indicator diagram from a cylinder fitted with trip motion and receiving boiler steam can always be distinguished by the sharp corner at cut-off.

The Reynolds Trip Gear.—As a typical example of trip motion the Reynolds gear is selected and illustrated by Fig. 90. This gear is the invention of Mr. Reynolds of the E. P. Allis

Company, Milwaukee, America, in which country it is most common. The patent right of the invention having expired any one is at liberty to manufacture the gear.

In Fig. 90 the double-ended lever A, to which the wrist-plate rod B is connected, swings loose upon the valve spindle C. This lever carries on its upper end a trip hook D, supported by a pin fixed in the lever A. The lever E is keyed on the valve spindle, and thus imparts any motion which it may receive to the valve. F is a square pin fixed in the lever E, and fitted with a steel catch or



Reynolds Trip Gear.

face as shown by the figure. On the lever E, and between the valve spindle C and the square pin F, another pin is provided. This pin G has a continual downward force exerted upon it by the dashpot through the medium of the dashpot-rod H, and will, therefore, unless prevented, always occupy its lowest position. A steel catch on the trip hook engages with the steel catch on the pin F; and while this engagement holds, any movement of the lever A in the direction of the arrow will draw up E through the medium of the trip hook, and E, being fast to the spindle, moves th valve. It will be seen that if by any means the catch on F becomes released

from the trip hook, the lever E will at once descend to its lowest position, and in so doing close the valve. It only remains to devise a means of effecting the disengagement at any desired instant, and the essentials of a trip gear are obtained. The method of accomplishing this disengagement is as follows:—The lever J, situated between the levers A and E, is loose upon the valve spindle. On the boss of this lever is a steel piece which projects above the surface of the boss. The inner limb of the trip hook rests upon this boss, being kept in position by the steel spring K encircling its upper half. During the motion of the wrist-plate lever A in the direction of the arrow, the trip hook is carried up, and because the steel catch thereon engages with the catch on the dashpot lever E, the latter also rises. This state of things will continue until the limb of the trip hook strikes the steel piece on the boss of lever J, when the hook is caused to oscillate upon its pin, and in so doing forces the steel catch clear of the catch on the dashpot lever, which is at once dragged down by the dashpot. From a consideration of the motion it will be seen that there is a sort of sharpening action on the catches when at work, whereby the edges remain in good condition for a considerable period. The steel catches are secured to their respective levers by small bolts as shown, and are sometimes designed so as to be reversible, thus presenting eight edges for wear.

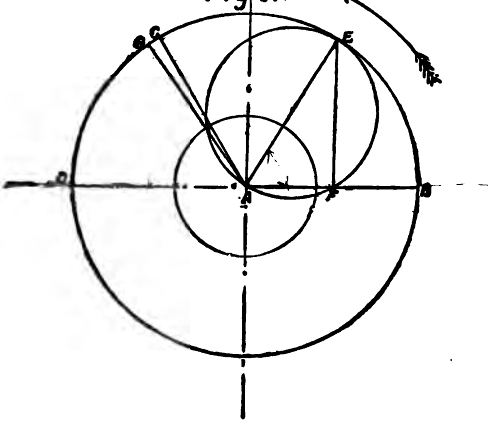
It is clear that by varying the angle of the trip lever the position of contact of the projecting piece thereon with the limb of the trip hook is also varied; and it is also clear that if J be connected to the governor in a suitable manner, the latter will have control over the cut-off by determining the point at which the lever E is released.

Diagram showing Range of Trip.—A study of Fig. 90 will reveal one important fact—namely, that tripping can only take place when the wrist-plate lever is moving in the opening direction. This statement is common to so many gears that it may be laid down as a rule.

Let A B, Fig. 91, represent the crank of an engine, and A C the eccentric having an angle the cosine of which is equal to lap *plus* lead. As drawn, the engine is on the back dead centre, and the steam valve at that end of the cylinder would be open to lead. The eccentric is moving the valve in the opening direction, and will continue to do so until it reaches A D. The angle of advance C A B is unaltered, and, therefore, when A C is at A D the crank is at A E, angle D A E being equal to angle C A B. The eccentric now returns, and unless the trip has operated before this return motion has commenced, it will not do so during that stroke. Therefore, with an eccentric set in the position A C, the total range of tripping is only from B to F, in this instance about 25 per cent. of the stroke. Such a range, however, is scarcely suitable for many cases, where it may be requisite on heavy loads to carry on steam to 50 or 60 per cent.;

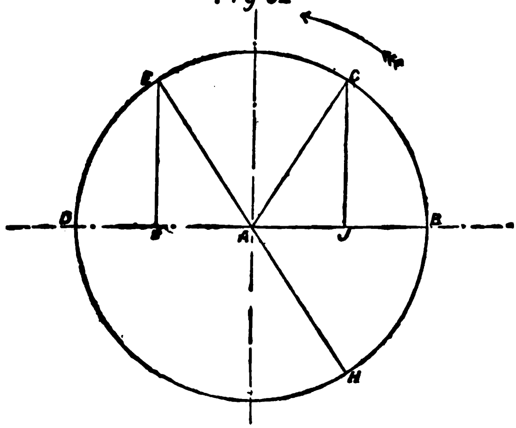
and at other times, when running light, to say 20 per cent. only. With the setting shown by Fig. 91, if the trip missed, steam would carry on till the crank reached A G, the valve then cutting off by virtue of its lap. Thus it is seen that between A E and A G the governor has no control over the steam.

Fig 91



Now take the case of an eccentric set in the position A C, Fig. 92. The letters in this diagram correspond with those in the previous figure. Here it is evident that the range of trip is considerably increased, being now B F, about 78 per cent. When

Fig 92



the crank is at B, the valve is open to lead as before, but when the eccentric is in this position, the crank will be at A H, angle H A B being equal to angle C A B. In order to give the desired amount of lead, the valve must cover the port by an amount equal

with the projecting piece just touching the trip hook, and this will be its position at latest trip.

The method by which the trip levers for each end of the cylinder are made to move equally in opposite directions is clearly shown in

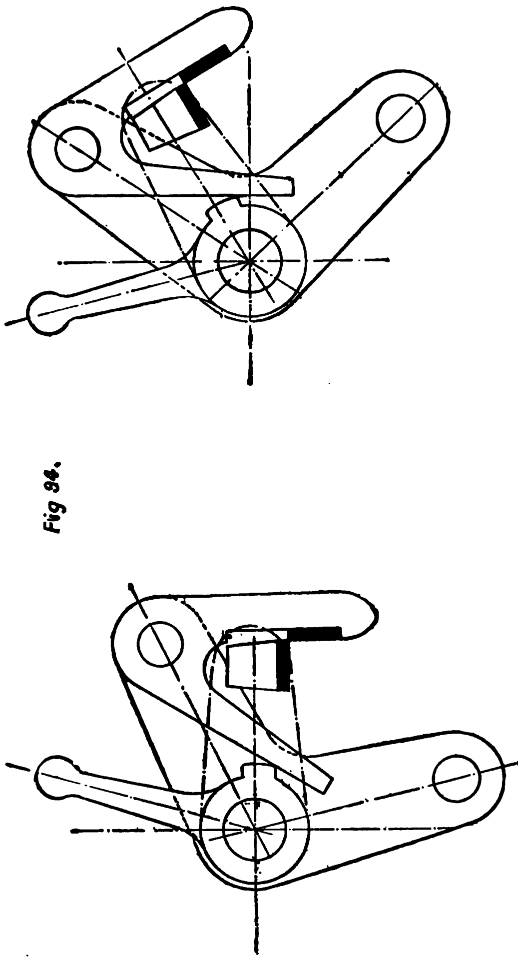


Fig 94.

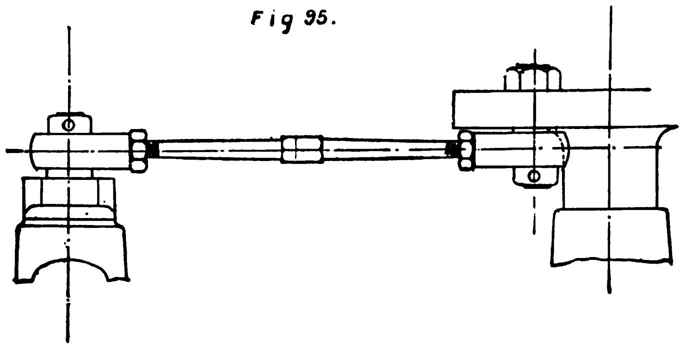
Movement of Trip Levers.

Fig. 93. With reference to this matter, it will suffice to say that P is a fixed centre of motion, and the governor actuates the double-ended lever swinging on centre P. A pointer may be attached to this lever playing over a graduated index, so that the point of cut-off is always seen.

Cause of Popularity of Corliss Gear in America.—In America, Mr. Corliss employed only one eccentric to operate both steam and exhaust valves; an arrangement whose very defect has perhaps contributed more to the popularity of Corliss gear than anything else. The one eccentric precludes the possibility of cutting off after about $\frac{4}{10}$ stroke, which was little later than the most economical point for pressures then common. Whenever a Corliss engine was put down, it was imperative that it should perform its work within this range of expansion, so that the owner was unable to run his engine except under economical conditions. The advantage over those engines which were too small for their work was at once apparent, and gave rise to a common notion that the economy was due solely to the valve gear.

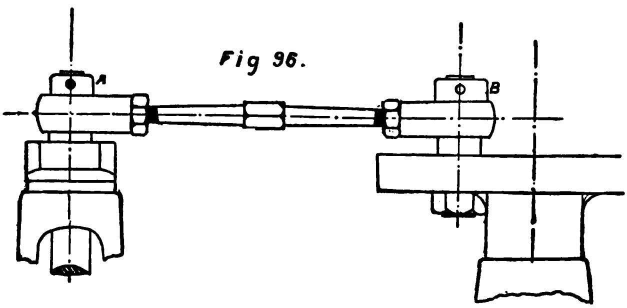
Arrangement of Link-work.—In designing the link-work of Corliss gears, it should be arranged that all rods can be taken off

Fig 95.



Arrangement of Link-work.

Fig 96.



Arrangement of Link-work (another method).

their pins by simply removing loose collars. This is a point sometimes forgotten, or at least neglected. In the arrangement shown

by Fig. 95, which is a plan view of a wrist-plate rod and its connections, it is evident that to remove the rod it is necessary to remove the rod from one of the sockets; whereas if it were arranged as shown by the next figure, the dismantling is effected by removing the two loose collars A and B. Then the length of the rod is not altered, and may therefore be replaced without any re-setting. These remarks apply to the governor trip rods, and in fact to all link-work of a similar nature.

CHAPTER III

DOUBLE-ECCENTRIC GEARS WITH TRIP MOTIONS.

THE advantage of double-eccentric gears has already been commented upon. It remains to discuss the action of this class.

Movement of Valves in Double-Eccentric Gears.—The difference between the motion of a single- and double-eccentric gear is, that whilst in the single-eccentric gear the dwelling period of the steam valves occupies half the travel of the wrist plate, the steam valves of a double-eccentric gear have, strictly speaking, no dwelling period; all the motion of the valve lever being taken up by lap, clearance of trip edges, and port opening. The steam valves never lap the port by an amount greater than the lap; hence, other things being equal, a less movement of the valve lever is required than with a single-eccentric gear by the amount of dwelling angle referred to the valve circumference. It has been common practice to make the augular movement of double-eccentric steam valves from 60° to 70°; and to make the port one-tenth the valve circumference. These are good rules, and, with an ordinary amount of lap, give sufficient over-travel to produce a full port opening early in the stroke.

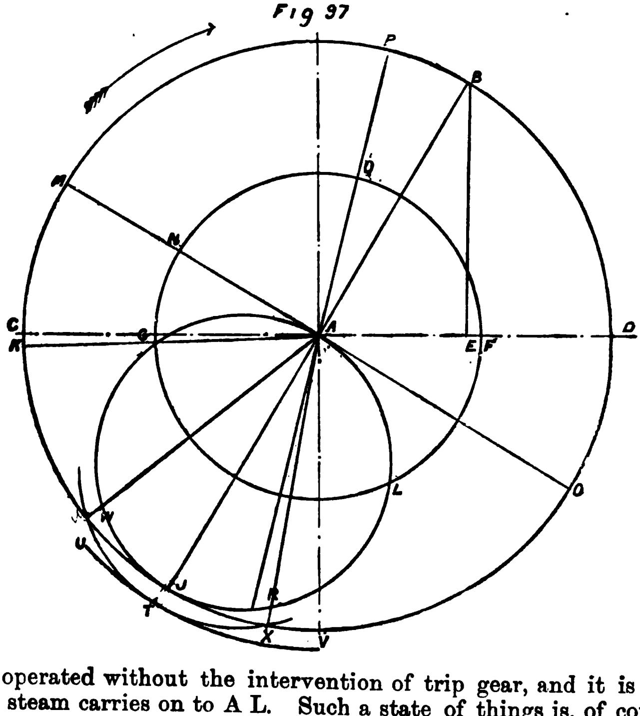
The action of the steam valves will be understood from the annexed diagram and the following description:—

Diagram of Motion.*—The circle B C D (Fig. 97) represents the travel of the valve by its diameter, and the path of the crank pin by its circumference. Let the total range of trip be from C to E—that is, about 75 per cent. Draw E B perpendicular to C D, and from E mark off the lead E F. Then, F D is the necessary lap of the valve, and the position of the eccentric is denoted by the line A B, the crank pin being, by supposition, at D. Describe the lap circle F N L. Produce A B to J, and describe the valve circle A G J. The valve and lap circles intersect at G. From A draw a line through G, and produce it to the crank-pin circle. Then A K is the crank position when admission commences. From this point

* This adaptation of the Zeuner diagram was first suggested by Mr. James Dunlop.

the port opening continues to increase, and at crank position A M is equal to A N. A N is at right angles to B J, and the port opening for any position of the crank between A M and A O, in the clockwise direction, is given by an ordinate between the lap circle and the remote side of the valve circle. Thus, at A P the port opening is Q R, and so on. After A O, the port opening is indicated by the ordinate between the lap circle and the adjacent side of the valve circle. The lap and valve circles intersect at L. This denotes that the port is closed, and the diagram shows that it remains so till A K is reached. This would be the steam distribution if the valves were positively connected to the eccentric-rod

Fig 97



and operated without the intervention of trip gear, and it is seen that steam carries on to A L. Such a state of things is, of course, absurd, but it must be remembered that the trip gear releases the catches before the crank passes A B, and the valve at once closes, remaining stationary until re-engagement takes place.

Speed of Trip Gears.—It will have been noticed that engagement of the catches must take place twice during each revolution of the engine, and as the catches in many cases only move past each other $\frac{3}{16}$ or $\frac{1}{4}$ of an inch, it will be understood that there is not much time for engagement. In fact, it is well known that a speed of 100 revolutions is about the limit for trip gears. When this speed

is exceeded, missing of the catches is a likely occurrence. This is serious in a single eccentric gear, because steam carries on far in the stroke, and regular running is impossible, but in double eccentric motions it would be positively dangerous, insomuch that a miss would cause live steam to flow through the exhaust, an occurrence not admissible under any circumstance.

In the previous figure let J T represent the clearance of the catches. With centre A and radius A T describe the arc U V. On A T as diameter, describe a portion of a circle cutting the valve travel circle in W and X. At X the catches are flush with each other. From X to T the clearance is being worked through, and at the latter point the valve lever is in its extreme position. From this point the catches approach one another and have re-engaged at W, at which point the valve begins to move. While the crank is passing from A X to A W the catches must re-engage or else a miss trip is the result. Take the case of an engine running at 100 revolutions per minute, and let the angle X A W be 30°. Under these conditions the time for re-engagement is only $\frac{60 \times 30}{100 \times 360}$; that is $\frac{1}{20}$ of a second.

It may be asked, why not give the catches more clearance to allow a longer time for engagement? The objection to this proposal is that more clearance would cause the engagement to occur at a time when the wrist-plate levers were moving at a comparatively high speed, and the inevitable result would be a considerable jar on the catches and wear on the pins. When the clearance is small, the eccentric is very near its dead centre when engagement takes place and is therefore moving the wrist-plate lever very slowly. The catches thus engage easily and without shock.

Range of Trip.—With respect to the range of trip that is most desirable, it may be stated that 75 per cent. is good and usual practice. If a less range will fulfil all requirements it may be adopted, but with the disadvantage of increased lap, which, to maintain the same port opening means increased travel, an objection which is compensated in some measure by full port opening being attained earlier in the stroke, as may be proved by construction. By increasing the range the lap is decreased, but the crank position at full port is later.

Proportions of Valves.—As there is no dwelling angle in the steam valves of double eccentric gears, it is sufficient to make the valve face lap the port on the back edge by an amount necessary for tightness and bearing surface when the valve is full closed.

The width of the face will therefore be—

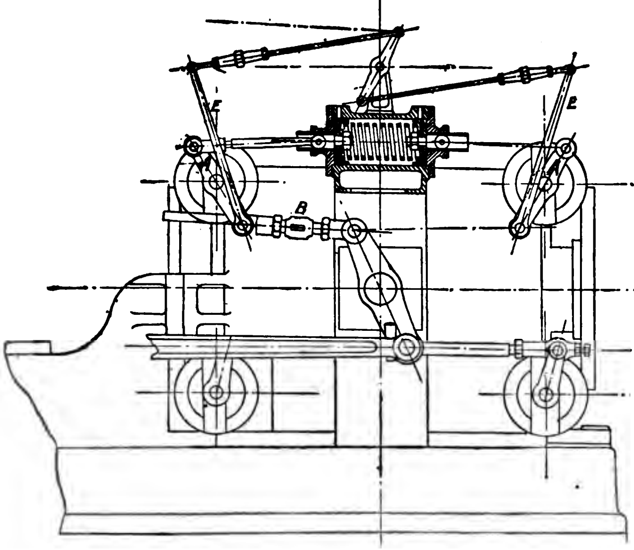
Steam lap + width of port + back lap.

The exhaust valves have dwelling periods, of course, like the single eccentric gears, the proportions for each case being exactly similar.

Messrs. Musgrave's Trip Gear.—The form of trip gear made

by Messrs. Musgrave, Bolton, consists of a valve lever A keyed to the valve spindle, and receiving motion from the rod B, which is provided with a steel catch. This catch engages with a steel piece

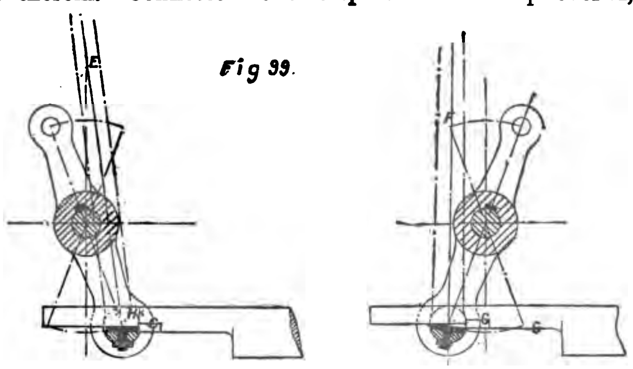
Fig 98.



Musgrave's Trip Gear.

on the valve-lever pin supported by the valve lever, but free to move therein. Connected to this spindle is the trip lever E, the

Fig 99.



Action of Catches in Musgrave's Trip Gear.

upper end of which is connected to the governor-rods in the manner indicated. The detail of the catches will be better seen in Fig. 99, in which the gear is shown in two positions when set for

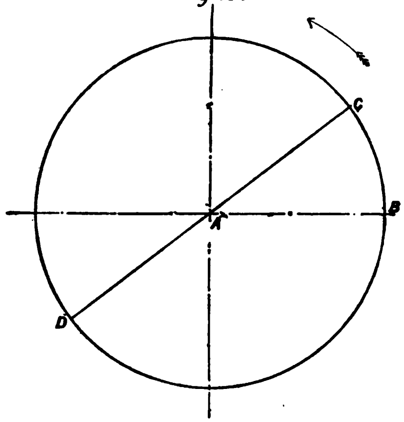
the latest trip. At the left hand the catches have just engaged, and the valve begins to move from right to left against the resistance of the dashpot spring. The valve lever continuing its motion causes the steel catch on the spindle to gradually become disengaged from the catch C, because the upper end is held by the governor, and for any one stroke may be considered stationary. In the diagram to the right the mechanism is at the extreme position in the opening direction. The catches are edge and edge and the valve lever is about to be moved by the dashpot, and to occupy the position indicated by the dotted line F G.

The length of the trip lever E is regulated by the depth of engagement, the travel of the valve lever, and the width of the catch on the trip lever.

Let H = distance from centre to edge of catch (see Fig. 99).
B = depth of engagement,
C = length of trip lever,
and D = travel of catch on trip lever,

Then, in order that trip may occur at the latest possible point, the equation $\frac{H}{C} = \frac{B}{D}$ must be satisfied. To find the depth of engagement at the beginning of the valve lever's motion in the opening direction, for any earlier point of cut-off, substitute D′ for D, D′

Fig 100



being the movement of the valve lever from the commencement of its travel to the point at which trip shall take place. The value of B for the earliest trip determines the movement of the end of the trip levers, it being equal to—

$$C\left(\frac{B - B'}{H}\right).$$

Here B′ is the depth of engagement or earliest trip. The whole

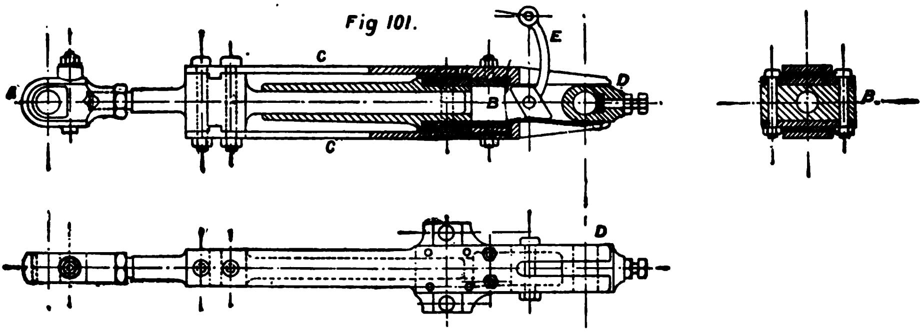
matter is simply a question of leverage, the length of the trip lever being one arm, and the half width of the catch plate the other.

If the travel of the valves is large, the trip levers become inconveniently long, and to avoid this, Messrs. Musgrave usually employ double-ported valves, thereby reducing the travel.

The arrangement just described is known as a "push" gear, because in opening the valve, the rod B is in compression. In push gears the eccentric must be directly opposite the proper position for a pull gear under the same conditions. Fig. 100 will illustrate the matter clearly. A B is the crank and A C the eccentric for a pull gear, but for a push gear A D would be the proper setting. The line A D is an extension of A B. It will be noticed that in Musgrave's gear the eccentric-rod connects to the wrist plate or its equivalent on the opposite side to which the valve-rods are coupled. This again alters the angle of the eccentric 180°, thus bringing it to A C, Fig. 100.

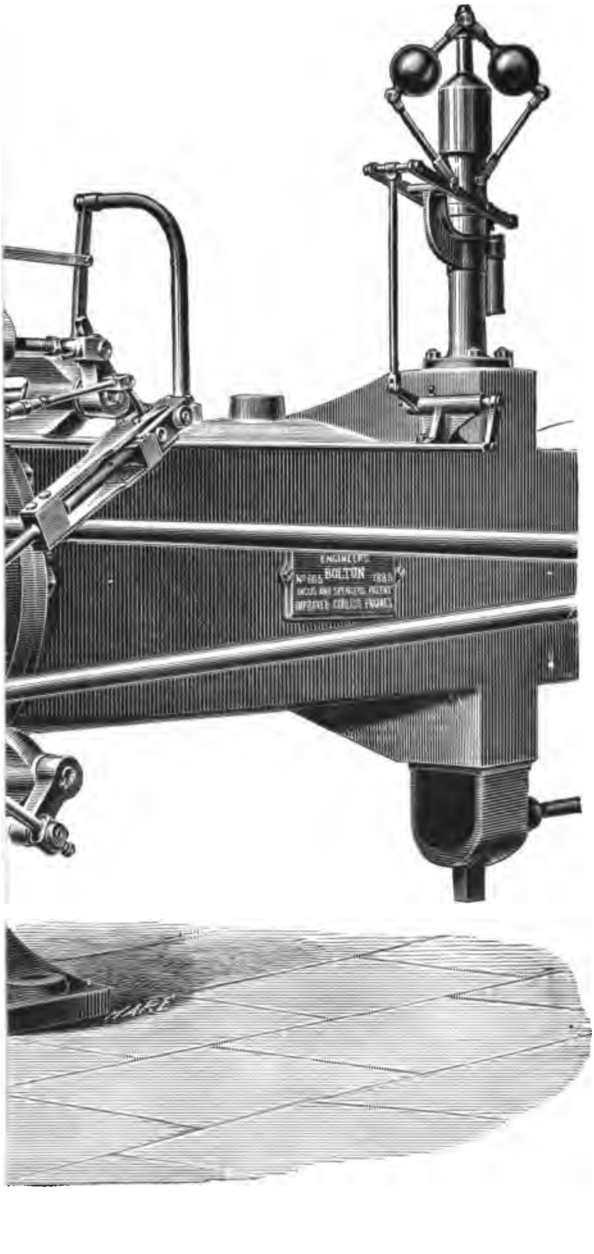
Inglis and Spencer Slip-Rod.—The well-known Inglis and Spencer slip-rod, the construction of which is shown in Fig. 101,

Fig 101.



Inglis and Spencer Slip-Rod.

has established itself as a simple and reliable form of trip gear. The end A embraces the wrist pin and carries the spring clips C C, which are secured by bolts as shown. The end B telescopes on this piece, and is provided with steel catches to engage with the pieces on the clips. The eye D is coupled to the valve lever. Tripping is effected by the toe lever E forcing open the steel clips and releasing the catches. The upper end of the toe lever is under the control of the governor, and by its inclination to the slip-rod determines the point of tripping. It will be seen that when the slip-rod moves in the direction from B to A and the upper end of the toe is stationary, the result is to cause the head of this lever to move round on its centre, and in so doing force open the clips. The piece B then springs forward in consequence of the pull of the dashpot on the opposite end of the valve lever to which B is connected, and thus the valve is closed. The part B is then stationary,


ENGINEERS
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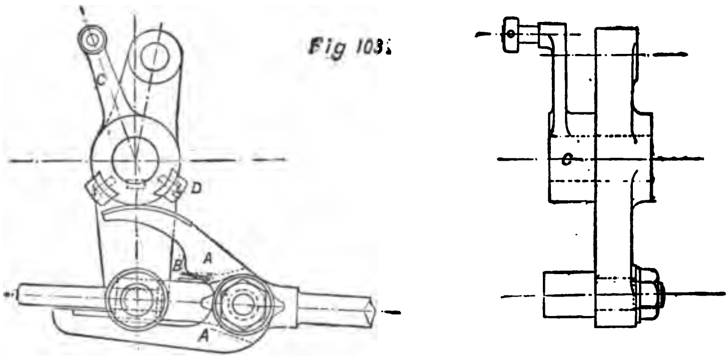
but A, being connected to the wrist plate, continues its motion until the catches on the clips move past the catches on B, so that engagement takes place, and the whole rod moves together. To prevent noise during engagement the clips are provided with leather pads, which, falling on B, prevent any metallic click that would otherwise occur.

Arrangement of Inglis and Spencer Trip Gear.—The annexed folding plate shows the practical application of the above form of trip gear. The makers are Messrs. Hick, Hargreaves & Co., Bolton. Separate eccentrics and wrist plates are employed for the steam and exhaust valves, thereby allowing of a wide range of trip. The Inglis and Spencer slip-rod is clearly shown, and also the connection of the toe levers to the governor. The trip-rods are made to move in equal and opposite directions by toothed sectors fastened to the spindles on which the levers carrying the trip-rods are secured. These sectors, which do not appear in the engraving, are situated under the dashpot. The gear is shown set for the latest trip, the back steam valve is wide open, and the toe lever is about to release the clips in the manner previously described. The exhaust portion of the gear calls for no special remarks, being the usual arrangement adopted when wrist plates are employed.

Crab-Claw Trip Motion.—The crab-claw trip motion originated in America, where it finds much favour. The double-armed



Crab-Claw Trip Motion.

valve lever carries a pin provided with a square head of hardened steel, which engages with the lever limb of the distinguishing feature of this gear—the crab claw A. The wrist-plate rod passes through the square head and is free to slide therein. The claw is centred on this rod, and within certain limits is free to move on its centre. It is held to its place by the small spring B. The action of the gear is as follows:—Suppose the gear to be in the extreme position and about to move the valve in the opening direction. The crab claw then catches the square head fixed in the valve lever and thus pulls the valve open. This action will continue till

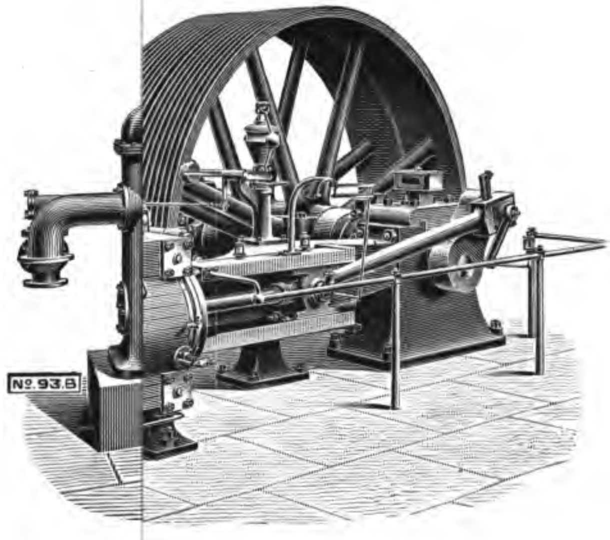
the upper limb of the claw strikes the projecting piece D on the governor lever C. The effect of this contact is to depress the crab-claw catch clear of the square head, and the valve lever being free, at once returns to its extreme position by the pull of the dashpot, and remains in that position until the crab claw returns to engage. The lever C is under the control of the governor, and by its position determines the time of liberation. The position of the crab claw centre for any point of the stroke being known, it is easy to find the position of D for any point of cut-off. As this matter was fully explained when describing Reynolds' trip gear, further explanation would be superfluous.

The second projection on the governor lever is for safety purposes. Should the governor driving-gear fail, the governor at once falls to its lowest position, and by moving the governor levers, brings the safety piece in contact with the claw and entirely prevents it from engaging with the square pin. The valve, therefore, remains closed, and the engine is soon brought to a standstill. The presence of the safety trip occasions a difficulty in starting; for the governor levers have to be held in a position to allow the claw to engage with the valve lever.

Messrs. Ruston Proctor's Crab-Claw Gear.—In England, Messrs. Ruston, Proctor & Co. employ the crab-claw gear, using separate eccentrics for steam and exhaust valves. Their practice is to carry the wrist-plate centre above the central line of the cylinder, if necessary, so as to bring the wrist-plate rods practically horizontal; and to locate the steam wrist-plate pins as near the vertical centre line of the wrist plate as the rod ends will permit.

Fig. 104 is an illustration of one of this firm's twin-compound condensing engines, fitted with crab-claw gear.

Dobson's Trip Gear.—Dobson's trip motion is illustrated by Fig. 105. The sliding piece A is carried by a bracket fixed to the side of the cylinder. This sliding piece receives its motion from the eccentric which couples to pin B. At each end of A are the trip boxes C, carrying tripping pieces E, and trip levers D. The latter are connected to each other and to the governor in the manner shown. At certain periods the tripping pieces engage with steel catches secured to the hollow spindle F, and by their engagement drag F and the dashpot piston secured thereto against the resistance of the dashpot spring. The upper ends of levers D are practically stationary, so that the effect of sliding A to and fro is to cause these levers to move on their centres, thus raising the tripping blocks and causing liberation. The cut shows the gear ready for tripping at about 25 per cent. of stroke. Any movement of the upper end of trip levers in an outward direction from the centre of the cylinder would delay liberation; whilst movement in an opposite direction would hasten it. The tripping blocks are held to their work by means of the light springs in the trip boxes, and are always pressed down upon the trip levers. The valve-rod



y Gear.

is of rectangular section at the middle to enable the end to follow the arc described by the valve lever pin.

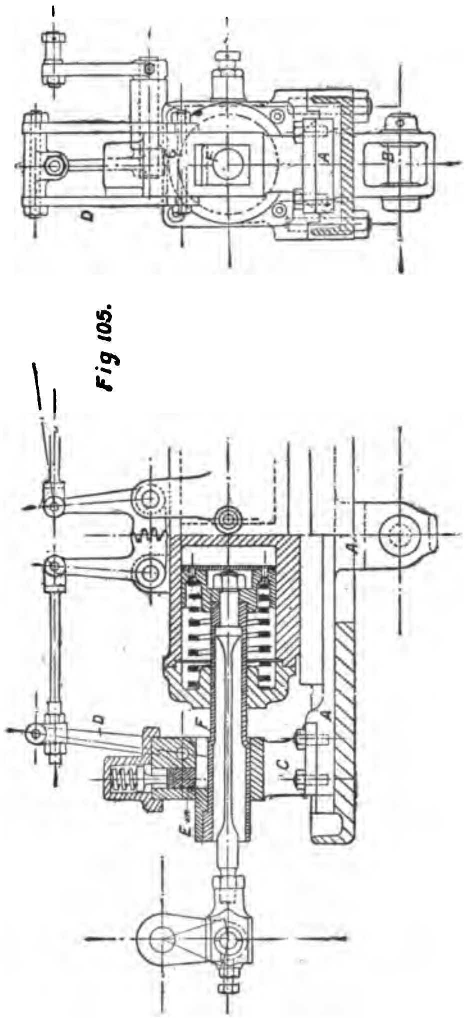


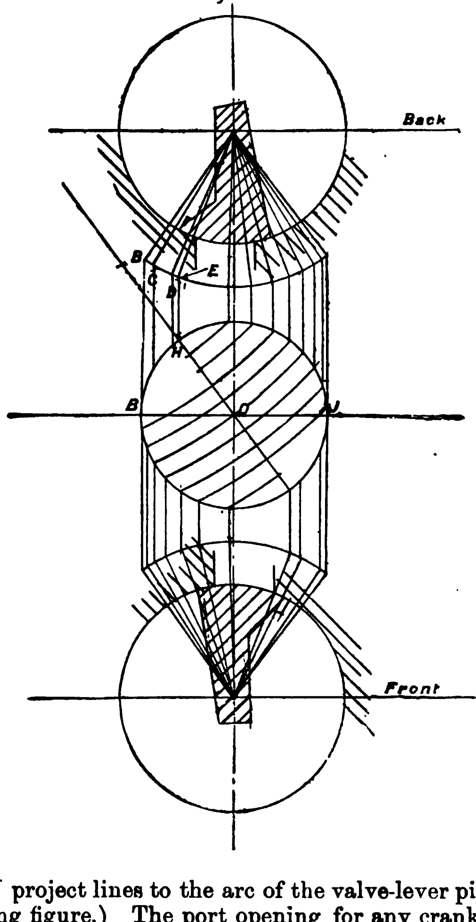
Fig 105.

Dobson's Trip Motion.

Valve Diagram for Same.—The valve diagram shows the motion of the steam valves for both back and front ends. Starting at point B (Fig. 106) for the back steam valve, the motion from B to C is taken up by clearance of trip catches. Lap is then worked off and the valve opens, and at D the admission commences. From D to E on the arc of the valve lever is lead, and the projection of E to

the circle indicates the position of the eccentric when crank is on dead centre. Let H be the projection of E, and let O H be produced indefinitely. With centre on O H produced, and with a radius which bears the same ratio to the radius of the circle B H J as the connecting-rod bears to the crank, describe arcs for any convenient number of crank positions. Where these arcs intersect

Fig 106.

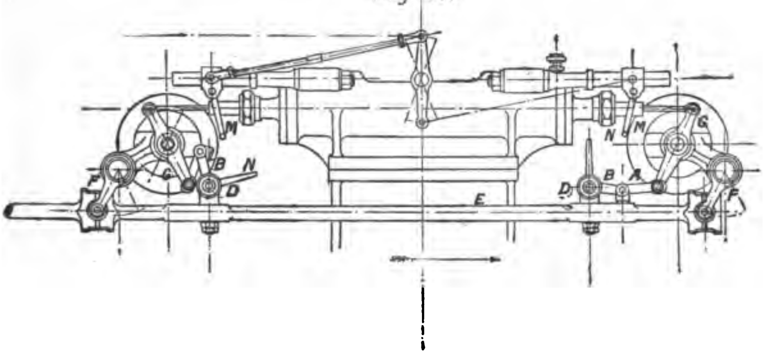


circle B H J project lines to the arc of the valve-lever pin. (See the accompanying figure.) The port opening for any crank position is then at once read off. In the present case, the range of trip is seen to be from J to H, about 75 per cent. The port opening at one-tenth stroke is 1½ inches; at two-tenths, 2¼ inches; at three-tenths, 2¾ inches; and so on. The unequal distribution due to connecting-rod influence causes the range of trip for the back stroke

to be somewhat less than for the front stroke. The diagram shows this. Messrs. Yates & Thom, Blackburn; Sharples & Co., Ramsbottom; and Victor Coates, Belfast, are constructors of Dobson's trip gear.

Goodfellow's Gear.—Benjamin Goodfellow, of Hyde, Manchester, constructs a trip motion in which catches are dispensed with altogether. In his arrangement, a knuckle joint is the device employed, and tripping is effected by the knuckle doubling up and leaving the valve to the influence of the dashpot. The rod E (Fig. 107) receives its motion from the eccentric-rod without the intervention of a wrist plate. The levers F F slung from brackets on the valve bonnets carry this rod, which is provided with a preparation for carrying a piece D. This piece forms the centre for the tripping finger. The double-ended valve levers C C are connected to the dashpot on one side, and to the piece D through the medium of the knuckle joint formed of the levers A and B. In the figure,

Fig 107.



Goodfellow's Gear.

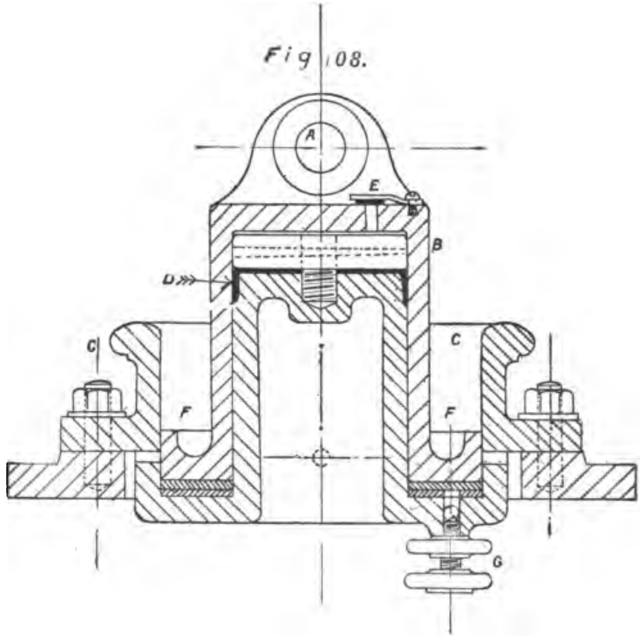
the gear is shown in the extreme position, and it will be seen that the knuckle for the back end is nearly straight. Any movement of E in the direction of the arrow will give an equal movement to the valve lever at the back end, because the central pin of the knuckle is slightly below the line connecting the two extreme pins, and the knuckle is prevented from doubling in a downward direction by a protuberance on the brass link A which rests upon the carrier rail E. The forward motion of the valve lever will continue until the tripping finger N, which forms part of the link B, comes in contact with the tripper M. When this occurs the knuckle at once closes up, and assumes a position similar to that shown at the front end. The trippers M are under the control of the governor, and by their position determine the point of cut-off. The hole in the brass link A is made larger than the pin of the valve lever which it embraces. The object of this clearance is to allow the centre joint of the knuckle to fall below the other joints, so as to

ensure no doubling up unless the trip finger is struck by the tripper. This play in the link corresponds to the clearance of ordinary trip gears. A leather pad on the link A deadens the noise when the gear attains its extreme position and the link falls on the carrier rail.

The method by which all plucking action on the governor due to the tripping lever striking the finger is avoided is very ingenious. The trip lever is hinged from the sliding collar, and is provided with stops on its upper end, which, whilst allowing the lever a little free play, prevent it swinging out of the way of the finger. The effect of M and N striking is not to slide the collar along the spindle and thus pluck the governor, but to cause the trip lever to bind on the under side of the sliding collar.

It will be convenient to conclude this chapter with a few remarks on dashpots.

Dashpots.—The vacuum dashpot shown by Fig. 108, rests upon a cast-iron plate secured to the engine foundations. The dashpot-

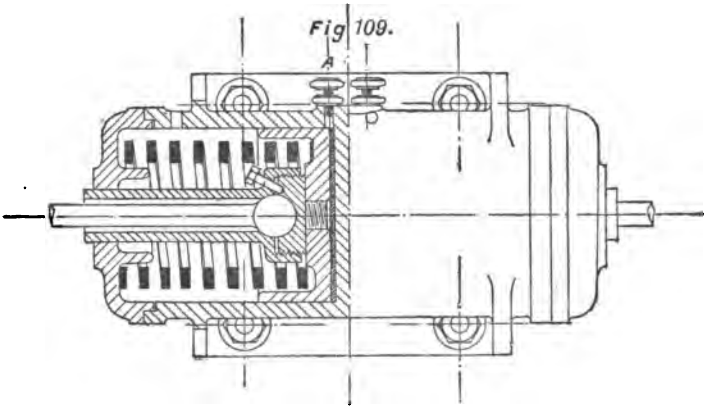


Vacuum Dashpot.

rod connects to the pin A, and lifts the plunger B, thus forming a vacuum between it and the casing C. Tightness is obtained by the leather ring D secured in position by a circular nut. The small valve E is free to move in one direction only, and allows air to escape from the vacuum chamber into the atmosphere. F is the

cushion chamber, the object of which is to prevent the dashpot closing violently. The cushion part of the plunger is provided with a leather pad which falls on a pad fastened to the bottom of the cushion chamber. When the piston descends, the small valve G regulates the escape of air from the cushion chamber with the greatest nicety, and renders the action practically noiseless.

The diagram of the spring dashpot is self-explanatory, and it only remains to say that the valve A regulates the cushioning of the piston in the manner already described.



Spring Dashpot.

A vacuum dashpot is not suitable for a double eccentric gear, because the small movement of the valve lever from engaging position to admission point does not raise the plunger sufficient to produce a good vacuum. There must be some clearance in the vacuum chamber, and therefore the degree of rarity of the atmosphere therein is proportional to the lift. In double eccentric gears the valve movement for early cut-off is very small, and is insufficient to secure prompt action of the dashpots; but with single eccentric gears, the dwelling and lap angles must always be worked through before steam opens, so that even at the earliest cut-off the dashpot lever lifts a considerable amount. With a spring dashpot, an initial tension or compression can be given, which will close the valves promptly at all grades of expansion.

CHAPTER IV.

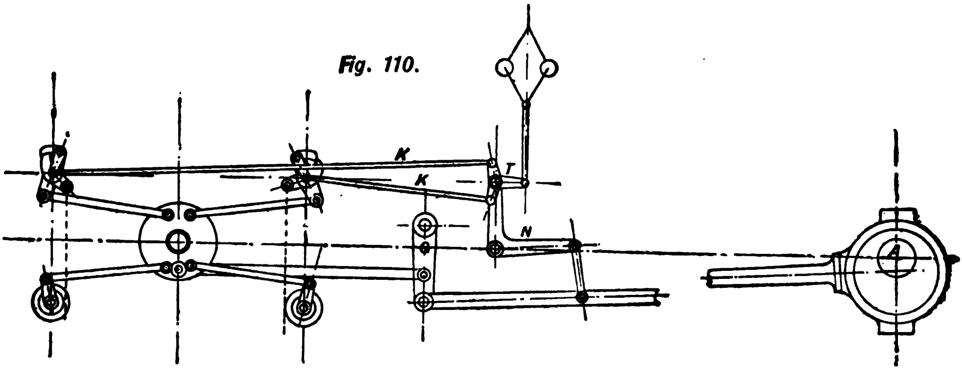
SINGLE ECCENTRIC GEARS WITH LARGE RANGE OF TRIP.

CONTENTS.—Frikart Single Eccentric Gear—Valve Diagram for Frikart Gear —Farcot Gear—Features of Cylinder—Wheelock Gear—Conclusion.

IT remains to illustrate and describe several gears which form exceptions to the rule that tripping can only take place when the eccentric is moving in the opening direction.

Frikart Single Eccentric Gear.—The first gear to notice is that invented by Frikart, and made under license by Messrs. Greenwood & Batley, Leeds. The peculiarities of this gear permit of

Fig. 110.



Frikart Single Eccentric Gear.

both steam and exhaust valves being operated by one eccentric, whilst giving a range of trip up to 75 per cent., or even later. Another advantage is that in consequence of engagement being positive—that is, not relying upon springs to bring the catches into gear—the motion is adapted to speeds quite unsuitable for ordinary trip gears.

In order to understand the action of the trip itself, it will be necessary to consider the arrangement of the rods and levers. In the diagrammatic sketch, Fig. 110, A is the crank shaft on which is keyed the eccentric, giving motion to the wrist plate through the swing lever G. The wrist plate communicates its motion to the exhaust valves in the usual manner; but the steam valves derive their motion through the medium of several levers. The detail of

the trip is shown by Figs. 111 and 112. The two levers A and B oscillate upon the valve spindle, the wrist-plate lever A being loose, and the dashpot lever B keyed on. Swinging on the upper end of A is the tripping lever and catch C, which, at certain periods,

Fig. 111.

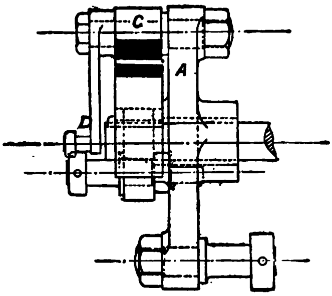
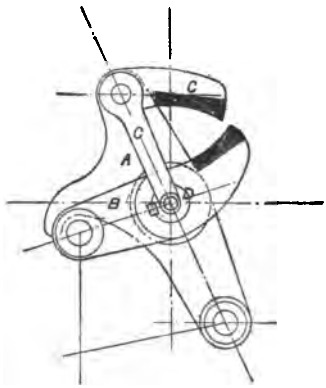


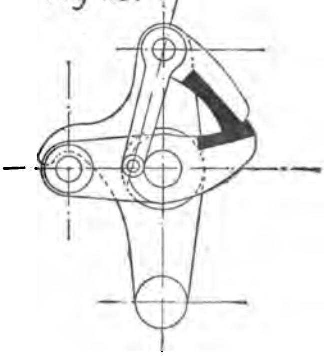
Fig. 112.



Frikart Single Eccentric Gear.

engages with the dashpot lever. C is so constructed, that when the centre of the pin D coincides with the centre line of the valve spindle, the dashpot lever is free to descend and cut off steam. Fig. 113 shows the gear in another position, where the centre of the trip lever pin is not coincident with the centre of the valve spindle; and where the dashpot lever is being raised because of the engagement of the trip lever with it.

Fig 113.



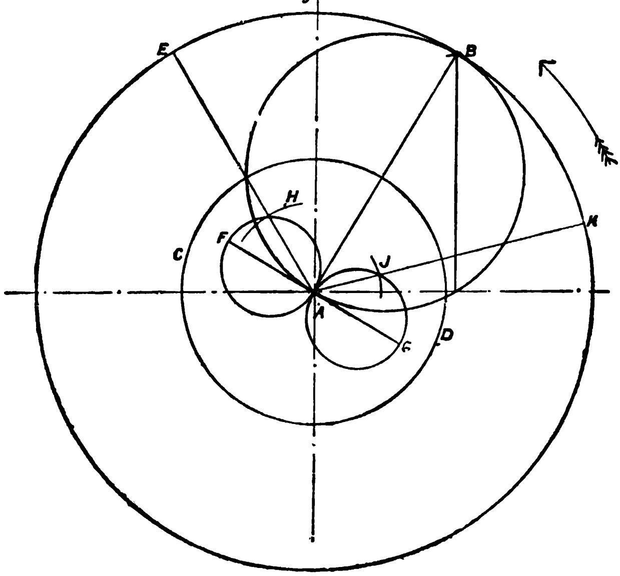
Reverting now to Fig. 110, the motion of the trip levers will be understood. From a suitable point on the eccentric-rod, the lever N receives motion. N in turn operates the levers K by means of the three-armed lever T. The governor connects to T by means of swinging links, and by its position determines the inclination of the three-armed lever. The latter has a constant motion from the eccentric-rod, so that the governor does not regulate the amount, but simply the position of that motion. The levers C are thus seen to be under the influence of three movements. In the first place they are carried round by the wrist-plate levers; the second motion is derived from the levers K; whilst the governor, as before stated,

regulates the position of the lever T, and, consequently, the time at which the trip lever pin concides with the centre of the valve spindle.

The lever A (Figs. 111 and 112) is so formed that in case of the dashpot sticking, the lever B will be forced to descend by coming in contact with the projecting part of the wrist-plate lever. With this arrangement there is no fear of the steam valve remaining open throughout the stroke, even if the dashpots fail to act.

Valve Diagram for Frikart Gear.—The motion of levers K is equivalent to that which would be derived from an eccentric set at right angles to the eccentric A, Figs. 111 and 112, and remembering this, it is easy to construct a diagram showing the action of

Fig 114.

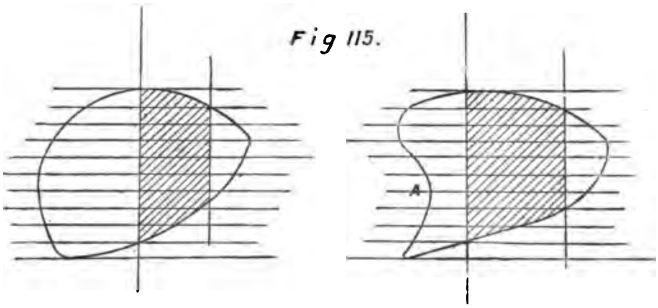


the trip levers very clearly. In Fig. 114, the position of the eccentric is given by the line A B. C D is the lap circle, and cut-off is therefore at A E. So far, the diagram is similar in every respect to the diagram for a slide valve, with the same lap and angle of advance. At right angles to A B, draw the line F G, equal to the travel of the trip levers due to the levers K; and on F G describe the circles as shown. When the governor is revolving in its normal plane, the coincidence of D (Figs. 111 and 112) with the valve spindle occurs when the crank is at A B, and cut-off is at this point. Suppose the governor to fall and move the centres of the trip-rods a certain amount. With this amount as radius, describe an arc cutting the polar circles at H. Produce A H,

which in this instance falls on the line A E. Tripping now takes place at A E, the point where the valve closes by virtue of its lap and movement from the wrist plate. This is, therefore, the latest trip. Suppose the governor rises and moves the trip levers in the opposite direction. Then the intersection of the circle whose radius equals the amount of displacement with the polar circles indicates the position of cut-off. In the diagram, A J represents the displacement, and tripping is seen to be at A K.*

Diagrams of the movement of the steam and exhaust valves are given by Fig. 115; that to the left being the steam diagram, and the right hand the exhaust. Referring to the steam diagram, it is seen that full port opening is attained at 10 per cent. in the stroke; a result which must be considered very satisfactory. The peculiar concavity of the exhaust diagram at A indicates that the disposition of the wrist-plate pins and exhaust

Fig 115.



valve-rods is faulty, insomuch that there is a double reversal of stress on these parts. The diagrams owe their irregular form to the presence of the wrist-plate, which modifies the motion so that the valve movement is no longer represented by an ellipse.

Farcot Gear.—The Farcot gear is another example of a single eccentric motion giving a wide range of trip. This gear was illustrated and described in *Engineering* some time ago, and the following description and cuts are taken from that source. (See Folding plate):—

"The general arrangement of the valve gear is shown by Fig. 116, while Figs. 117 to 120 represent the cut-off gear in detail. As will be seen from the general view, but one eccentric is used, this giving motion to a wrist-plate, which is connected by rods *c* to levers *d* mounted loose on the admission valve spindles (see Figs. 117, 118, and 119). The lower part of each lever *d* carries a catch *f* which is constantly drawn towards the valve spindle by a spring. Upon the valve spindle is keyed an arm *g*, which is acted upon by a spring tending constantly to close the valve, and which, on the other side of the valve spindle, carries a steel catch plate *h* (Fig.

* This diagram is due to Mr. James Dunlop.

118), arranged so as to engage with the corresponding steel plate *j* carried by the catch *f*. It will be readily understood how by this arrangement the oscillation of the lever *d* in one direction gives an opening movement to the valves, while the closure of the latter under the pull of the spring connected to the arm *g* will evidently take place whenever the steel plates *j* and *h* are disengaged from each other.

"The disengagement of the catches *h j* is effected, under the control of the governor, by an ingenious arrangement, which we shall now describe, and which permits of the cut-off being varied from zero to 80 per cent. of the stroke. In the ordinary Corliss gear it will be remembered that the detachment of the valve must take place while the latter is opening, and thus, if detachment has not taken place before the valve is fully open, the admission goes on to the full extent permitted by the undetached valve. In the Farcot gear, on the other hand, the detachment may take place on either the opening or closing strokes of the valve, the means by which it is effected being as follows:—On the boss through which each admission valve spindle passes, are mounted two cams *k k'*, as shown by Figs. 117, 119, and 120; these cams being coupled by the links *l l'* to one arm of a bell-crank (see Figs. 116 and 117), the other arm of which is connected to the governor. The projections on these cams act upon the detent *f* as follows:—The projection of the cam *k* acts directly on the finger *m* (Fig. 120) and serves to lift the detent *f* and release the valve during the opening stroke of the latter, thus cutting off the steam at any point from zero up to about 35 per cent. of the stroke; the cam *k'*, on the other hand, acts upon the movable finger *n*, and serves to release the valve during its closing stroke, thus cutting off the steam at from about 35 to 80 per cent. of the stroke.

"Referring to Fig. 120, it will be seen that the finger *n* slides within *m*, and is pressed out by a spiral spring. During the opening stroke of the valve, *n* is pressed within *m* by its end coming into contact with an inclined surface on the cam *k'* giving the longer admissions. In this way the cam *k'* is prevented from interfering with the action of the cam *k* when the engine is working with short steam admissions. On the other hand, when the opening movement of the valve has taken place sufficiently far, the projection on *k'* completely passes the finger *n*, and the latter is then forced out by its spring ready to engage with the projection of the cam *k'* during the return stroke of the valve.

"The wearing parts of the gear we have been describing are of simple form, and are readily renewable, while, being made of hardened steel, the wear on them is exceedingly small. As compared with the Farcot gear exhibited in 1878, that now described has the advantage that the arrangement for giving late admissions is not in continual use. When working with the short admissions usual with this class of engine, the finger *n* remains projecting freely from *m*, and it is only thrust back within the latter when longer admissions

are required. The slopes of the cams k and k' also being very gentle, exceedingly little work is thrown upon the governor.

"A special arrangement of one of the cams also prevents the engine from running away in the event of the governor breaking down, this cam shutting off the steam completely in the event of the governor stopping and the balls closing. It will be noticed on reference to our detail views that the valve spindles are without stuffing boxes, they simply working in well-fitting gun-metal bushes fitted to the bosses through which they pass."

Features of Cylinder.—A section of the cylinder is shown by Fig. 121. The endeavour to reduce clearance by placing the valves in the covers will be noticed and appreciated. The steam-jacketing is a special feature of this engine; and an inspection of the figure will show that the covers are also constructed to give ample space for boiler steam.

A point to which exception may be taken is the double reversal of stress on both steam and exhaust rods and wrist-plate pins, due to the particular location of the latter; but this point has evidently been subordinated to that of keeping the dwelling angles as small as possible. That there is a double reversal of stress on these parts is apparent from the general view of the engine.

Wheelock Gear.—As another example of a single eccentric gear giving a wide range of trip, the Wheelock gear may be cited. The rods which actuate the trip levers receive their motion from the eccentric in a similar manner to the Frikart gear. The governor determines the position of the motion of the tripping piece, but not the amount, that being constant at all times. Strictly speaking, the valves are not of the Corliss type, but consist of multiple-ported grid valves working on a flat face. This face is formed on a conical plug of special hard metal forced into suitable taper holes in the cylinder. The connection of the spindles with the valves is through the medium of a toggle joint, situated inside the valve chamber, a device which has the advantage of giving a rapid opening motion; a good feature usually purchased at the expense of a wrist plate. Messrs. Daniel Adamson & Co., Dukinfield, are manufacturers of the Wheelock gear.

Conclusion.—There are numerous other varieties of trip gears, but as they are all pretty much on the same lines as those already described, it will be unnecessary to describe them. A knowledge of a few good and well-known types will enable the action of others to be understood, and their merits and demerits appreciated. Although many makers seem resolutely to adopt one style of gear, and use it to the exclusion of all others, the advantage of one gear over another is but slight, and it may be taken that, given good design and workmanship, it is immaterial which gear is selected, because the above qualities ensure satisfactory operation to most types.

Extensive as is the use of Corliss gear, it has not yet been employed in all cases where it would be advantageous. Marine

engineers, although striving in many ways to reduce weight and steam consumption, have so far neglected the Corliss valve, but if further economy is to be looked for, the Corliss valve undoubtedly offers a means of attaining it; and with the ever upward tendency of steam pressures its advantages will become more pronounced. The chief difficulty with which the marine engineer will have to contend will probably be the arranging of the gear in a compact and accessible position, so that the valves may be drawn out for inspection, without unshipping more parts than is absolutely necessary. Trip gear, of course, will be dispensed with, the valves having lap equal to a slide valve under similar conditions. Joy's radial valve gear, or some other similar contrivance, would perhaps be preferable to link motion, for reasons which will be apparent from a comparison of the two arrangements.

PART III.

DOUBLE-BEAT VALVES AND MISCELLANEOUS GEARS.

PART III.

DOUBLE-BEAT VALVES AND MISCELLANEOUS GEARS.

CONTENTS.—Cornish Valves—Valve Motion for Cornish Valves—Form of Cams—Cornish Valves with Trip Motion—Robey's Valve Gear—Sulzer Valve Gear—Beadle's Valve Gear—Duplex Pump Valve Gear.

The Cornish or Double-Beat Valve has not been extensively adopted in this country, except by makers of large winding and pumping engines; but in a modified form it has for years been largely employed by Continental builders, by whom it has been brought to such perfection that it already compares favourably with the Corliss valve gear. The future practice in steam engines promises to utilise the double-beat valve very largely; for, with the revival of superheating, arises the want of a valve that has no rubbing action on its seat. A similar change took place in the design of gas engine valves some years ago. In the old engines the cycle of operations was controlled by slides, whereas the modern engines have mushroom valves.

Dealing first with the valve proper, the object of the designer should be to produce a mechanism that is as short as possible, light, and has the least possible lift. The common form of Cornish valve is shown by Fig. 122. The narrower the faces or beats the easier is the valve kept tight, but if too narrow they soon become worn and require renewal. The lift necessary to give the full area of the lower seat is (neglecting the wings of the valve)—

$$\tfrac{1}{4}\left(\frac{D^2}{D + d}\right),$$

where D and d are the diameters of the lower and upper seats respectively. The wings or ribs which are necessary to prevent distortion of the valve under the steam pressure diminish the area, consequently a little more lift is required than is given above, but at the same time the area through the lower seat is considerably diminished by the ribs and the central boss, this should be taken into account from the first, and allowance made, otherwise it may be that the design may be far advanced before the insufficiency of the area is first discovered.

A = width of bar for lower seat.
B = 2 lift + A.
The area of the annulus C = half the area of lower seat.

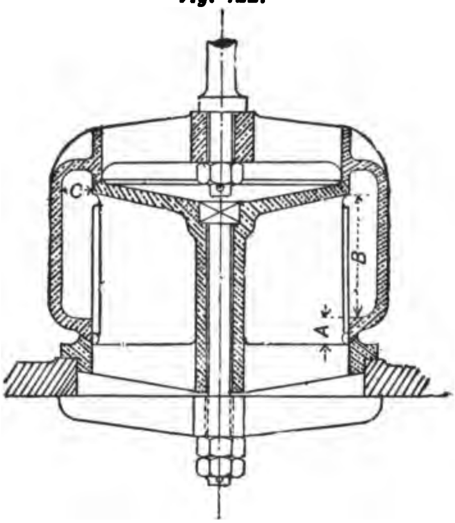
Unless the distance between the two faces be made as short as

possible, the valve may be leaky, owing to the difference between the expansion of the valve and the seat. The upper seating plate should be well ribbed, or the steam pressure will deflect it, and leakage will ensue; and the valve and seat should be cast of metals having equal coefficients of expansion. With respect to the valve faces the following widths have been found to give satisfaction:—

Valve 6	inches diameter	inside	largest seat,	$\frac{1}{8}$	inch	width of seat.
" 8	"	"	"	$\frac{3}{16}$	"	"
" 10	"	"	"	$\frac{5}{16}$	"	"
" 12	"	"	"	$\frac{3}{8}$	"	"
" 14	"	"	"	$\frac{1}{2}$	"	"

The angle of the faces in each case was 45°, and the width given is measured horizontally, hence the actual width of the seat is $\sqrt{2W^2}$. It will be seen on inspecting the figure that the width

Fig. 122.

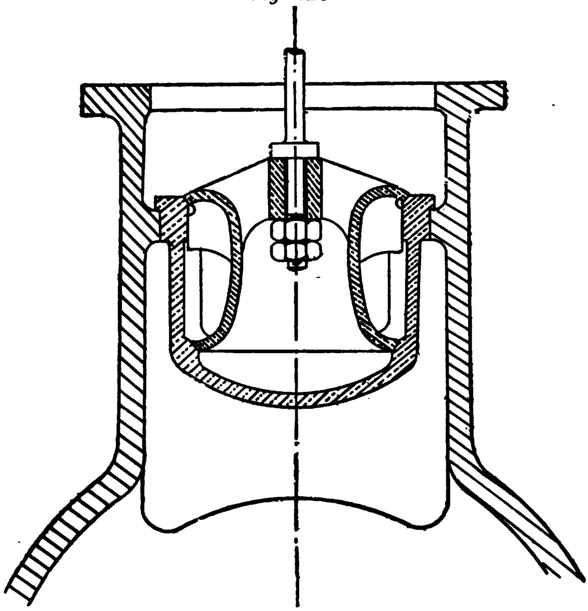


of the faces controls the difference in the diameters of the upper and lower seats, and consequently the power required to lift the valve off the seat. Were the two diameters equal, the valve would be in equilibrium except for its own weight, and the unbalanced area of the spindle, which might counterbalance each other; but unless the valve were under the control of a spring it would be leaky, and the presence of any grit on the faces would seriously impair its working. When the difference between the diameters of the two faces is, however, sufficient to give the widths above mentioned, sufficient unbalanced pressure is brought on the valve to keep it tight, and hammer away the small amount of foreign

matter that might, under the ordinary conditions of working, be deposited on the valve faces.

In the next figure a more modern form of double-beat valve is illustrated. This is an improvement on the previous design, because for a given area the valve is lighter. In all valves of this type it is important that both valve and seat should be of the same metal, so as to avoid difference between the expansion of the two parts, otherwise there will be a continual leakage of steam.

Fig. 128.



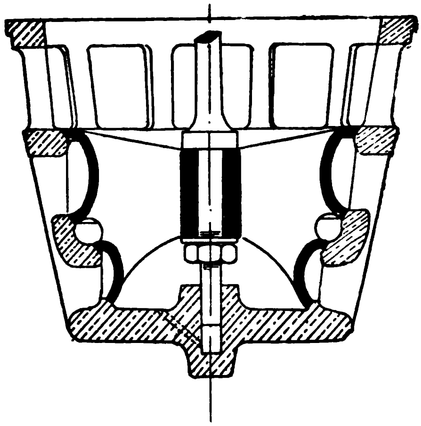
By combining two double-beat valves an extremely small lift for a given area is obtained, a point of some importance in large engines where trip gear is used. Such a valve is shown in the annexed figure. With this type the necessity of attention to equal expansion is emphasised, there being four faces liable to leakage; and first-class workmanship is indispensable to satisfactory working.

A common arrangement of Cornish valve motion is shown in outline in Fig. 125. The position of the cams depends upon the desired point of cut-off. The valves, of course, can have no lap, but by placing the steam cams so that they do not lift the valve until the valve-rod is past its central position the same effect is produced as by a lap on a slide valve. If the levers and cams were set so that when the valve-rod was in the centre of its travel all the valves were just on the point of being lifted or released, the

resulting distribution would be similar to that given by a slide valve having neither steam nor exhaust lap.

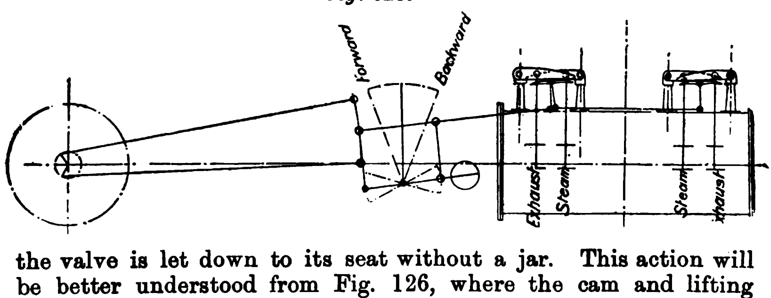
In designing the cams it is important to see that the first point of contact is near the fulcrum of the cam. The advantage of this

Fig. 124.



is twofold. It throws less strain on the valve gear, because at the instant of contact the valve is not in equilibrium. In the second place the speed of the striking part of the cam is slow, and the action, therefore, quiet. When the valve has been lifted it is in equilibrium, and as the point of contact is continually receding from the fulcrum of the cam, the opening is rapid; also, on closing,

Fig. 125.

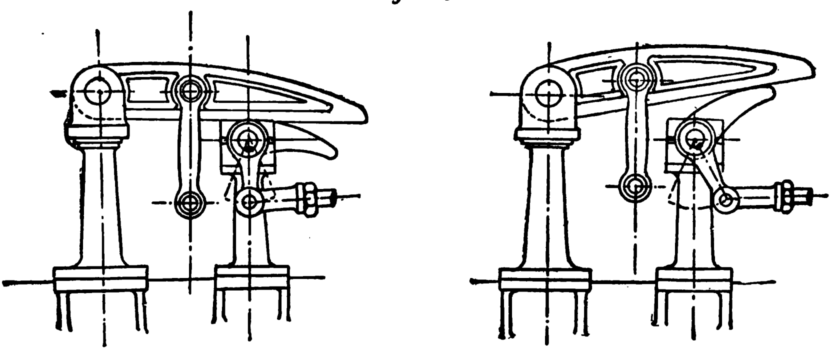


the valve is let down to its seat without a jar. This action will be better understood from Fig. 126, where the cam and lifting lever are shown in two positions, the first in which the valve is on the point of being lifted from its seat, the second in which the valve has attained its extreme lift.

The Cornish valve, when adapted to a tripping arrangement,

forms a useful gear for large winding engines. The essential feature of this class of engine is that it must start rapidly and maintain its speed to the latest possible point in the wind. The gear about to be described accomplishes the desired action in a very effective manner. It was designed by the author in 1898 for a large winding engine in the Lancashire coal-field. In Fig. 127 the valve-rod A and the rocking-rod B receive motion from the eccentrics through the medium of a reversing motion of the straight link type. Until the maximum speed is attained the steam is carried on for nearly the whole length of the stroke, but when the maximum is reached the governor moves the small cams C into a position that brings

Fig. 126.



them in contact with the rising trip hook. The valve, spindle, stirrup, and hook are then left to fall to the closed position by reason of their weight, and the air cushion, which can be regulated by a small thumb screw, prevents the valve falling violently on its face. In cases where the valve is light and the pressure high, a spring would be necessary to close it promptly, because of the unbalanced area of the valve spindle and the friction of the stuffing box. The maximum speed being once attained, a very small amount of steam is sufficient to maintain it, because the load becomes lighter every instant, owing to the diminishing weight of rope on the ascending cage and the increasing weight on the descending rope. Tripping, of course, must take place before the trip hook begins to descend—that is, somewhat before half-stroke —as in the single eccentric Corliss gear. The indicator diagram from a complete wind will illustrate the action clearly.

From the figure it appears that after about five revolutions the speed at which tripping occurs is obtained. For several strokes the speed increases, as shown by the early tripping, until, towards the end of the wind, a mere breath is admitted. The stop valve is then closed, the brakes applied, and the engine rapidly brought to rest.

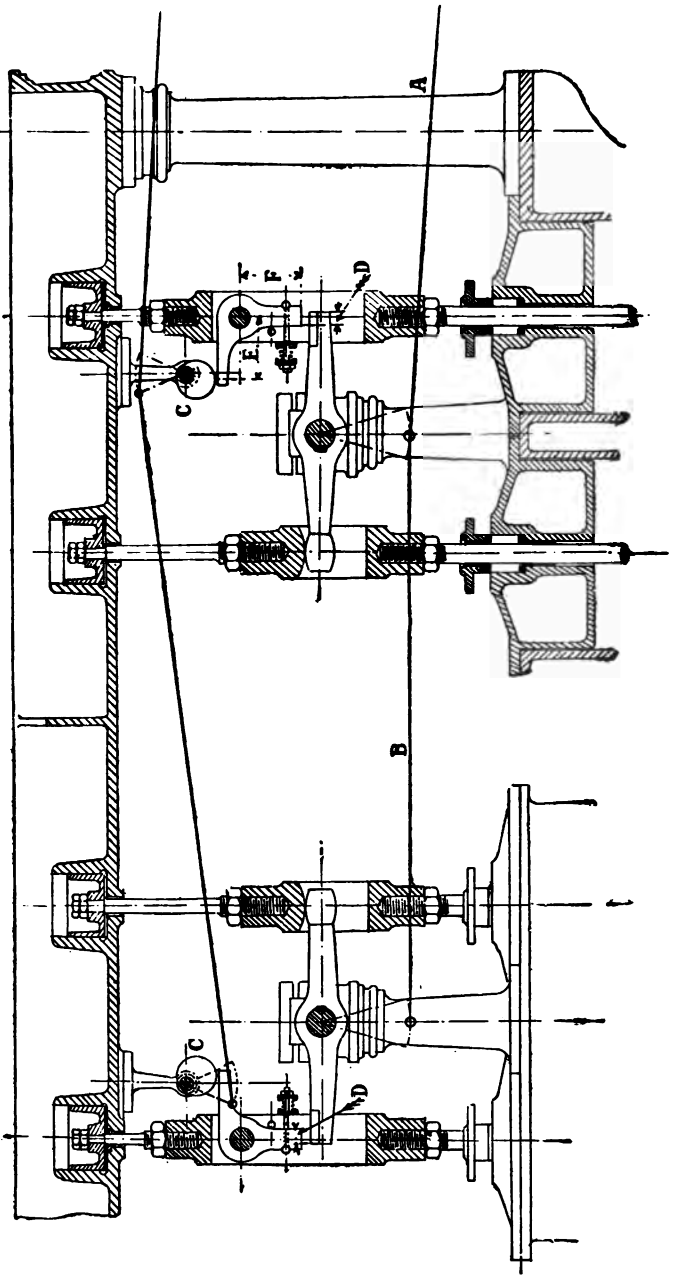
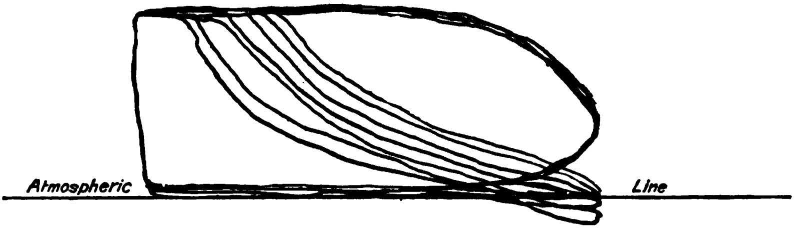


Fig. 127.

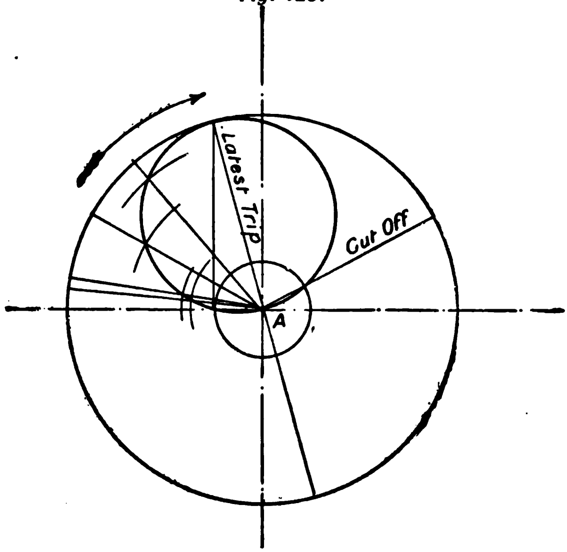
The action of the trip may be clearly shown by means of a Zeuner diagram. In the figure the clearance between the lifting

Fig. 128.



lever and the trip hook is drawn as a lap circle, and the lead being fixed upon the angle of the eccentric is known. Draw the valve circle, and from the centre A, with a radius equal to the distance

Fig. 129.



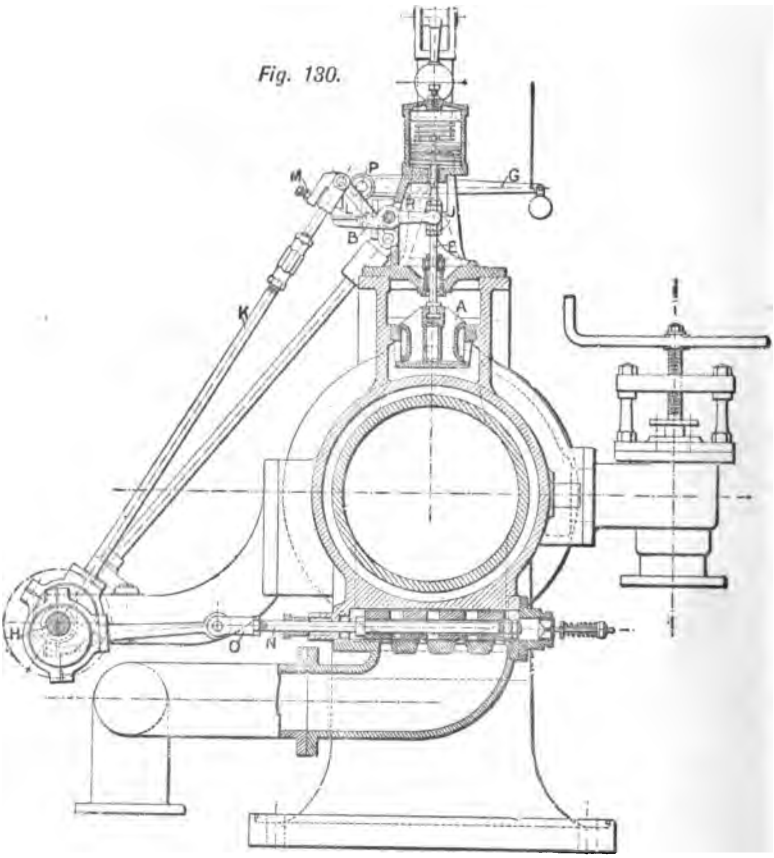
between the pallett of the catch and the cam, + clearance between lifting lever and trip hook, + overlap of tripping edges (D in Fig. 127), describe an arc intersecting the valve circle.* A line drawn

* This construction applies where the distance from the fulcrum of the catch to the point at which the cam strikes is equal to the distance from the fulcrum to the working edge of the catch. Should these be unequal the radius of the arc will be increased or diminished proportionally. In Fig. 127, should the distance E exceed the distance F, the radius of the arc will be greater by the difference of these distances referred to the distance D.

from A through the point of intersection will show the crank position when disengagement takes place. Should the arc fall without the valve circle no disengagement will take place, and steam will be carried on until cut-off occurs by virtue of the clearance of the catches.

In non-reversing engines the gear for a double-beat valve motion is more conveniently arranged. A horizontal shaft driven by bevel or skew gear is carried parallel to the centre line of the engine. On this shaft the eccentrics are placed at convenient positions. The best known English example of this type of gear is that made by Messrs. Robey & Co., Lincoln. The arrangement will be followed from the accompanying figures. The method of governing

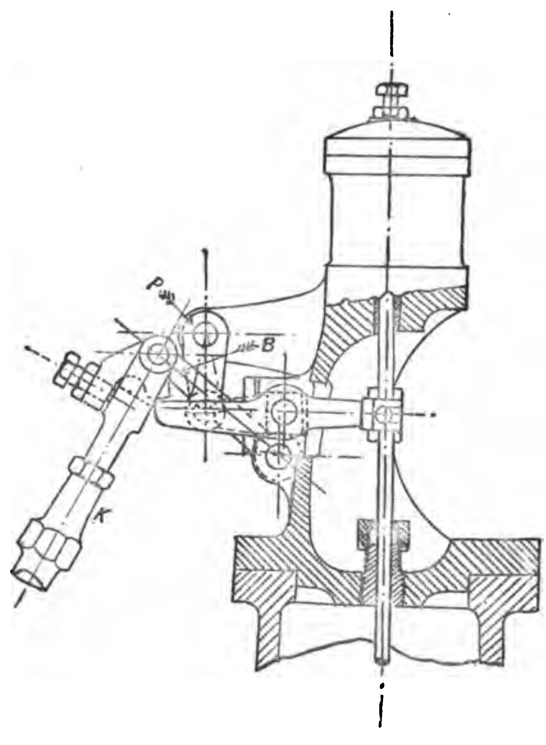
Fig. 130.



and tripping is similar to that of an ordinary Corliss gear, and the range of tripping depends upon the clearance of the catches and the

lap of same, just as in the Corliss gear the range depends on the ratio of the lap of the valve to its travel. The steam eccentrics are placed at an angle of 180° to each other—that is, opposite. The exhaust valves, it will be noticed, consist of gridiron slides at the bottom of the cylinder. The advantages of this arrangement are the excellent natural drainage of the cylinder and the short travel of the slides. As in the case of the steam valves, the exhaust eccentrics are set opposite to each other. Returning to the detail

Fig. 131

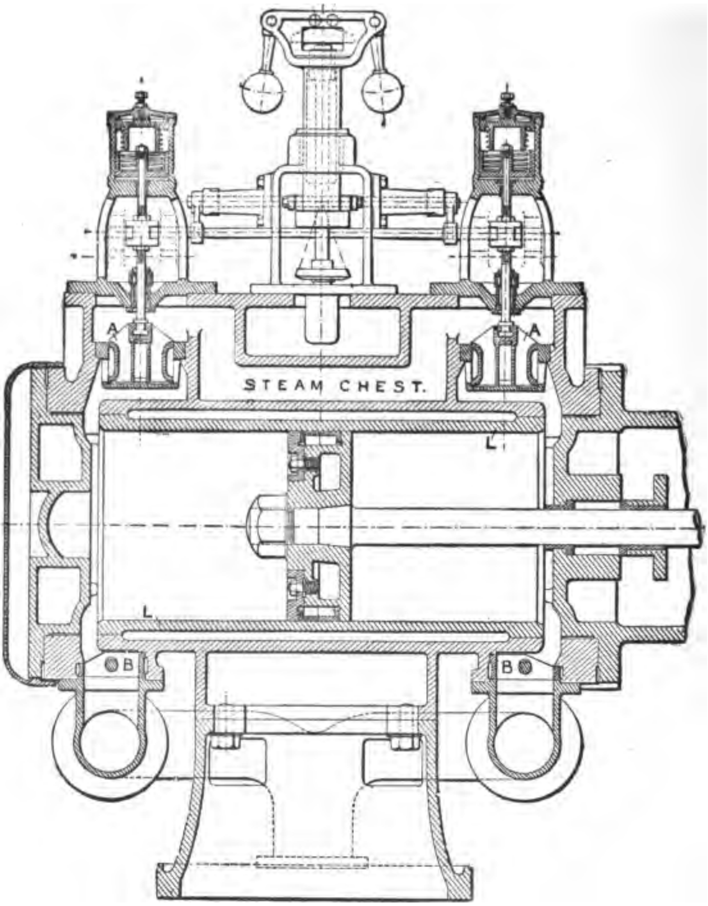


ot the trip motion (Fig. 131) the end of the eccentric rod K is carried by the anchor levers B, the fulcra of which are carried by the valve bonnet. As previously stated, the point at which tripping takes place depends on the depth of engagement of the catches, which depth can be adjusted in two ways—permanently by altering the set screw on the catch lever, and temporarily by the action of the governor on the lever P, thus altering the position of the lifting lever. The set screw adjustment is resorted to when

any permanent change in the load is made, or when a permanent increase or decrease of speed is desired; whilst the governor adjusts the cut-off so that the desired speed is uniform.

On setting out the movements of this gear it will be found that a very small movement of the lifting lever will produce a wide

Fig. 132.



variation in the point of cut-off. This means that the mechanical purchase between the balls of the governor and the lever P (Fig. 131) is considerable, consequently the plucking action on the governor is very slight. The lifting lever is forked where it

engages with the valve spindle, and the movable fulcrum is carried in hardened slides, thus permitting the necessary horizontal movement.

The well-known **Sulzer gear** is one of the most interesting examples of drop valve gear, the latest type having several features of great interest. The folding plate illustrates the valve gear on the high-pressure cylinder of the engine exhibited at Paris in 1901. At first sight the gear appears to be unreasonably complicated, but a careful inspection of the figure will show that every part contributes to the efficiency of the gear, and nothing that is wanted in a high-class gear has been forgotten. A general idea of the action will be formed from the plate, but a thorough knowledge of the various movements can only be obtained by drawing down the centre lines and tracing their movements through a complete revolution. Commencing at the lay shaft A, the eccentric-rod B imparts motion to the bell-crank lever C, one of whose arms actuates the exhaust gear, whilst the vertical arm controls the opening of the steam valve. The connection between this arm and the valve-rod D is not of a positive character, however, but arises through the medium of catches E. The fulcrum of the curved lever F G is at P, and as long as the catches are in gear the revolution of the valve shaft in the direction of the arrow will, until the eccentric-rod begins to return, drag down the valve-rod D and lift the steam valve. The fulcrum P also serves as an anchor for the catch lever. This lever likewise receives its motion from an eccentric, and the rod Q connected thereto receives a positive and constant amount, but the position of the motion is altered by the governor operating upon the horizontal shaft H. This portion of the gear resembles the Frikart gear in its action. In the drawing the normal position of the governor lever is shown. As the lever J moves nearer the vertical centre line the release of the catches will be delayed. On the other hand, movement in the opposite direction will accelerate release.

The chief point of interest in this gear, however, lies in the rolling valve levers and adjustable palletts. Referring to the steam rolling lever K, and assuming it to be in its highest position, the first point of contact is close to the end of the lever, which means that the gear has a great leverage over the valves just at the moment the stress comes upon the motion. As soon as the valve is off its seat it is practically in equilibrium, and the rolling lever is then continually decreasing its purchase, thus imparting rapid opening to the valve at the time the piston is travelling at high speed. As to the pallett, it is centred on the pin M, and is adjustable by means of nuts on a thread. The action of the main dashpot will be readily followed, but the supplementary cushion spring is a refinement which Messrs. Sulzer find necessary to give a proper closure to the valve, and its action is peculiar. In the drawing the steam valve is shown full open, but it will be

understood that this is not its correct position in relation to the position of the rest of the gear, but it is so drawn in order to show the extremely short lift of valve. The break in the valve stem indicates that the valve and gear are not drawn in their true positions relative to one another. Returning to the cushion dash-pot, it will be observed that when the catches are released the main spring descends with the valve, at the same time lifting the valve rod D, the curved lever, and the cushion dashpot piston. There is no cushioning action on the main spring, but as the cushion piston approaches the end of the cylinder a break is brought upon the main spring, and as this acts at a purchase it is possible to obtain an exceedingly delicate adjustment. Had only the main spring been cushioned some means would have had to be provided for limiting the motion of the rolling lever and the rods in connection with it, otherwise the descent of the main spring would jerk them badly, hence the break cylinders become vital parts of the gear. It is, of course, highly important that the regulation of the closure of the valves should be entirely under control; any hammer on the faces would soon destroy them.

In the exhaust gear the rolling lever and pallett are employed, and the valve is opened against the resistance of a spring, but there is no release gear or cushioning. The valve is shown full open, but that is not its correct position with reference to the rest of the gear.

The plate is well worth close inspection. The very complete arrangements for lubrication are specially interesting, as well as the precautions against waste oil running upon the foundations. The lagging question has also been thoroughly thought out, and accounts in a large measure for the splendid appearance of the Sulzer engines. Another point which is worth notice is the absence of over-hung pins. All levers are jawed where they embrace the rod-ends, or else the rod-ends are jawed and embrace the levers. This renders the manufacture of the moving parts costly, but Messrs. Sulzer ignore expense where efficiency of the gear is concerned.

Beadle's Rotary Valve Gear.—The requirements of an ideal valve gear are perfect equilibrium, unlimited speed, and small clearances. The rotary valve gear made by Mr. C. Hyde Beadle realises to a considerable extent these requirements, and is also very compact. It is most conveniently made with two cylinders, either compound or otherwise, with pistons half a stroke apart, or any other desired position. It will be convenient to illustrate and describe the "helicrank" engine, because it embodies some features which render it of particular value for marine purposes, the cylinders being in axial alignment with the tail shaft.

Referring to the accompanying figures, four views of the helicrank engine are shown—a vertical section through the valve box, a sectional plan through the plane of the piston-rods, and two

end elevations of the engine with the valve chest covers removed. The ends of the piston-rods A are provided with adjustable rollers B, which roll upon the helix C, thereby imparting a rotary motion

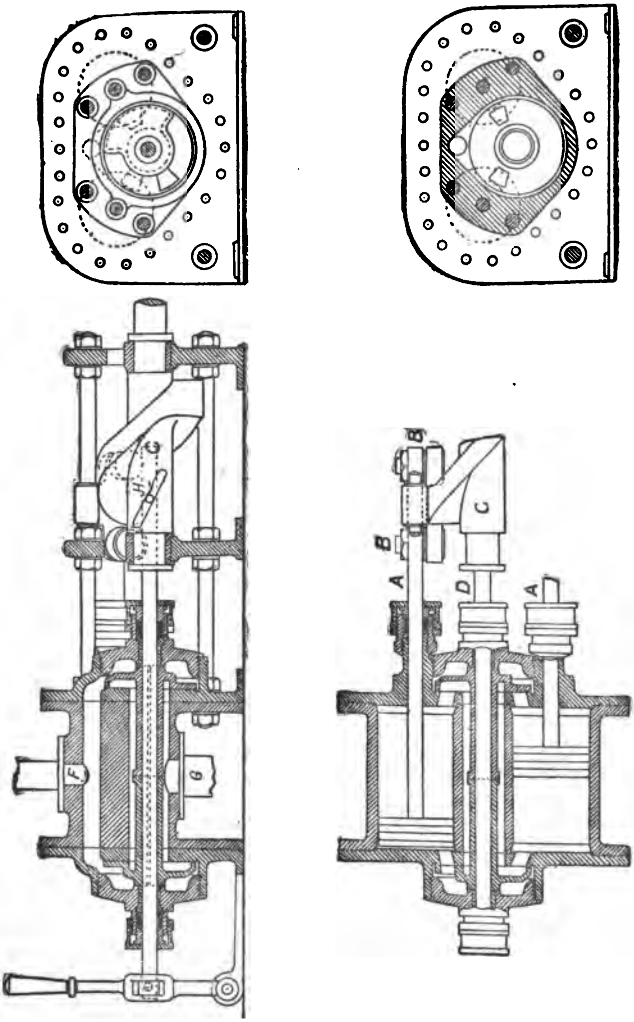


Fig. 184.

to the main shaft, and also to the valve spindle D. The valves consist of two revolving discs with prolonged stems, and suitable slots and cavities to effect the proper distribution. The valves are

mounted on the spindle by means of feather keys, whereby they always rotate with the spindle, but, at the same time, are free endways. The faces of the stems are accurately faced, and form the stop which prevents the valves from being forced against their faces. The force tending to press one valve to its face is exactly balanced by the force on the back of the other, and consequently the valves float, rather than rub, upon their face. The steam enters at F, from thence it circulates round the cylinders to the valve chests. The extremely short and direct ports on the cylinder will be noted, and also the absence of external pipes or passages. The exhaust passes round the prolonged stems of the valves, and escapes through G.

The reversal of the rotary valve engines is a very simple matter. One method is that shown in the figure. The valve spindle fits into the helix as shown, and is driven by the peg H. The reversing gear is shown in the mid position, and the action of the valves is such that no rotation would be given to the main shaft, but when the reversing handle is but moved into any other position the effect is to impart axial motion to the valves through the inclined slot in the helicrank. The effect of notching up is almost identical with that of notching up a link motion, only the action is more harmonic.

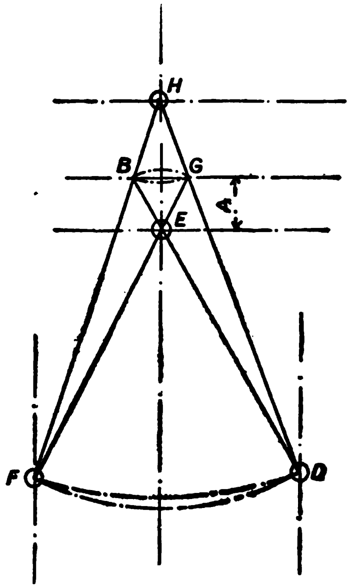
As showing the easy running of these valves, it may be mentioned that after running six months the valves of one of these engines still exhibited the minute original scratches made by the scraping tool. In ordinary crank engines the valve spindles are driven by mitre or skew gear from the crank shaft in a similar manner to the shaft of a gas engine. Any desired sequence or angle of crank is accommodated by a suitable disposition of the ports leading to the cylinders and the design of the valves themselves.

With crank at right angles, or with a helicrank engine with one piston half a stroke in advance of the other, lines joining the centres of the cylinders form a right-angled triangle, and one slot in the valve thus distributes the steam to both cylinders, and one recess controls the exhaust.

Duplex Pump Valve Gear.—Fig. 136 shows a section through one of the steam cylinders of a duplex pump. The general arrangement of this type of pump is well known. There are two distinct and independent cylinders placed opposite and in axial alignment with two pumps, and the piston-rod of each cylinder is coupled directly to the pump-rod. The valve gear of one cylinder is operated from the crosshead of the other. There are two ports in each end of the cylinder, of which the outer ports B are provided for the entering steam, and the inner ports A for the exhaust. When in central position the valve exactly covers the two ports as shown in the illustration. The travel of the valve should not be less than twice the width of one port, and it is often convenient to make it greater. The peculiarity in the connection of the spindle to the valve will be noticed. The nuts are adjusted so that there

is an amount of lost motion between the spindle and the valve, and in arranging the valve levers this must be taken into account. It is impossible, however, to predetermine the exact amount of this lost motion, hence it is the usual practice to provide for adjustment by an extension of the screwed part of the spindle. Usually the total amount of lost motion varies from $\frac{1}{8}$ inch for pumps with steam cylinders of 4 inches diameter to $\frac{1}{2}$ inch in cylinders 24 inches diameter and upwards. The effect of having too much lost motion is to give a long stroke to the piston, and since there is no mechanical limit to the stroke except that provided by the cylinder covers, the piston may strike. Too little lost motion gives short strokes, which diminishes the delivery of the plungers. As the piston approaches the end of the stroke the inner port is blocked up, and the steam remaining in the cylinder at the front of the advancing piston forms a cushion, which brings it gently to rest. Sometimes cushion valves E E are provided, by adjusting which the motion of the piston may be further controlled. From the foregoing it will be understood that the arrangement of the cylinder end must differ from an ordinary cylinder. In the first place, the bell-mouth must not extend past the inner edge of the steam port, otherwise cushioning cannot be efficient, because there would be leakage through to the exhaust port. The ports should also be arranged as shown in the figure, and not recessed into the cover as is common practice in ordinary cylinders, otherwise it will be difficult to obtain proper cushioning, because the piston would strike the covers almost immediately after the inner port was blocked by the piston, and until the port has been covered cushioning cannot begin.

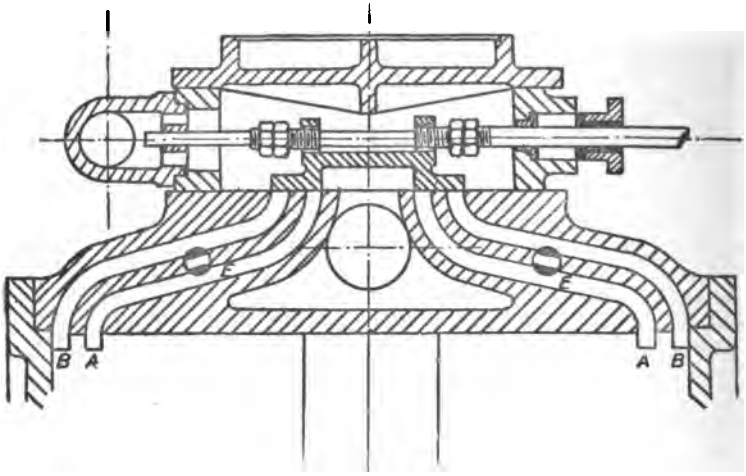
Fig. 136.



Passing on to the design of the valve levers, each valve has to effect the same travel. The cross-shafts are usually arranged above each other, and, therefore, vary according to the lengths of the levers, but the movement of each extremity must be equal to that of the other, and the proportion or ratio between the valve lever and crosshead lever must be the same in each case. Another point to observe is that the length of the shortest valve lever is sufficient

to allow of the connecting link to the valve spindle clearing the cross shaft. Referring to the diagram (Fig. 135), the stroke of the crosshead is from F to D, and is a known quantity. The travel of the valve spindle B G can also be ascertained. The minimum distance A is fixed by the detail of the connecting link and the size of the cross shaft. From B or G draw a line through E cutting the verticals indicating the length of the stroke. The point of intersection indicates the position on the crosshead where the connection to the crosshead lever must be made, and also the length

Fig. 136.



of the lever. Should the position be inconveniently high, a lower point may be selected and the distance A left to find itself. A higher point must not be chosen if A has already been made a minimum. Having determined the proportions of one crosshead and valve lever, draw lines from F or D through B or G, and the point where they intersect the vertical line is the proper centre of the cross shaft. F H or D H is the length of the crosshead lever, and B H or G H the length of the valve lever, and the two valve travels are the same.

Sometimes the lost motion adjustment is outside the steam chest, especially in the case of large sizes. This enables the working of the pumps to be adjusted under steam.

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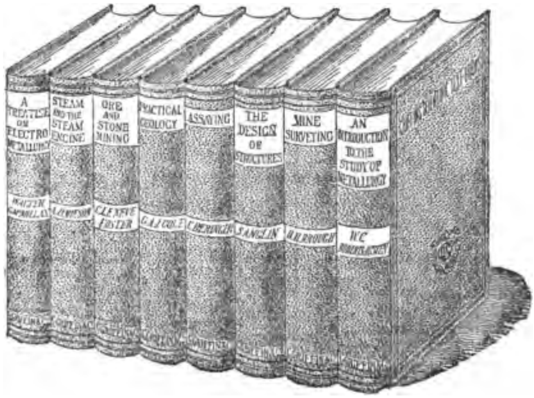

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BY CHARLES DIXON.

The Spacious Air.—The Open Fields and Downs.—In the Hedgerows.—On Open Heath and Moor.—On the Mountains.—Amongst the Evergreens.—Copse and Woodland.—By Stream and Pool.—The Sandy Wastes and Mud-flats.—Sea-laved Rocks.—Birds of the Cities.—INDEX.